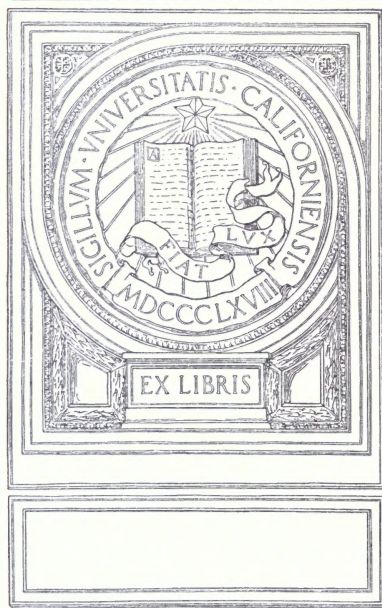
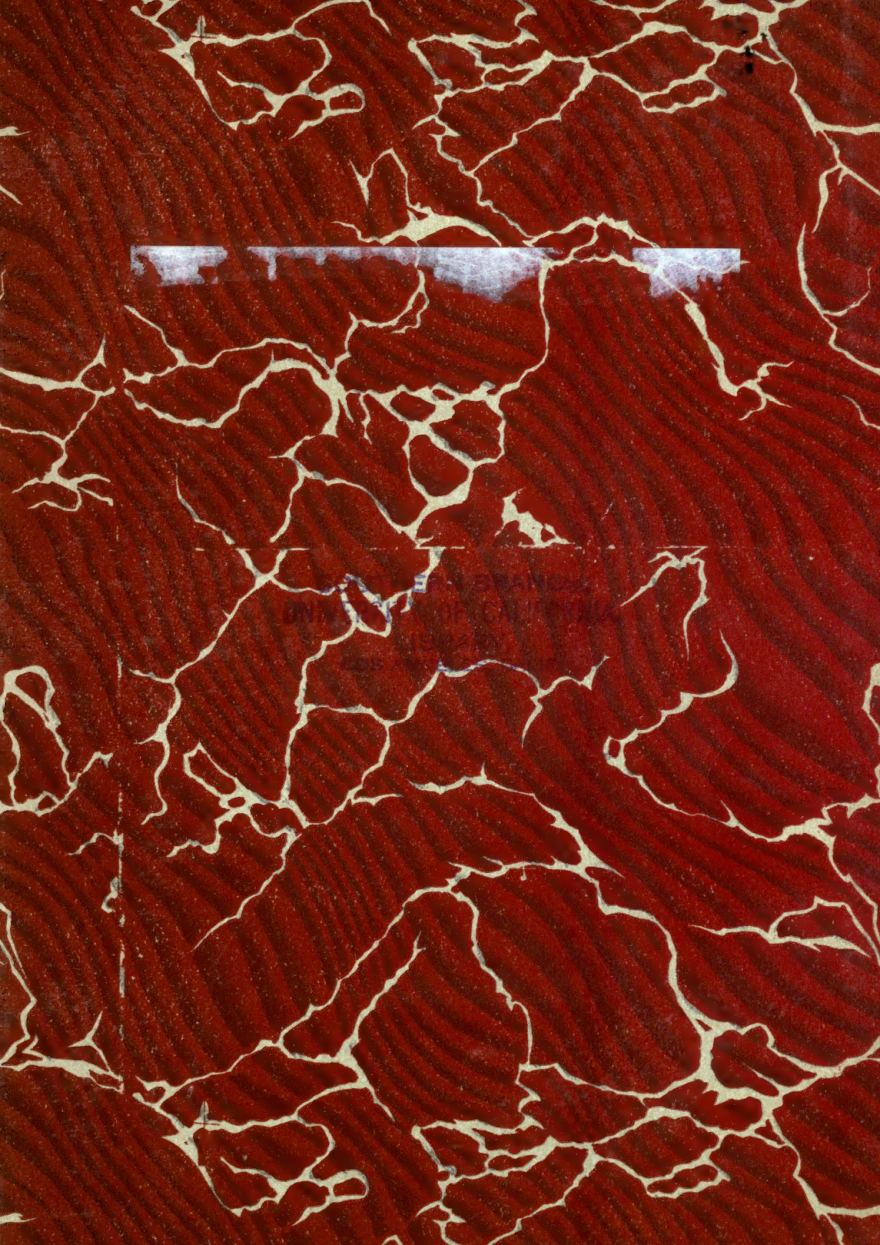


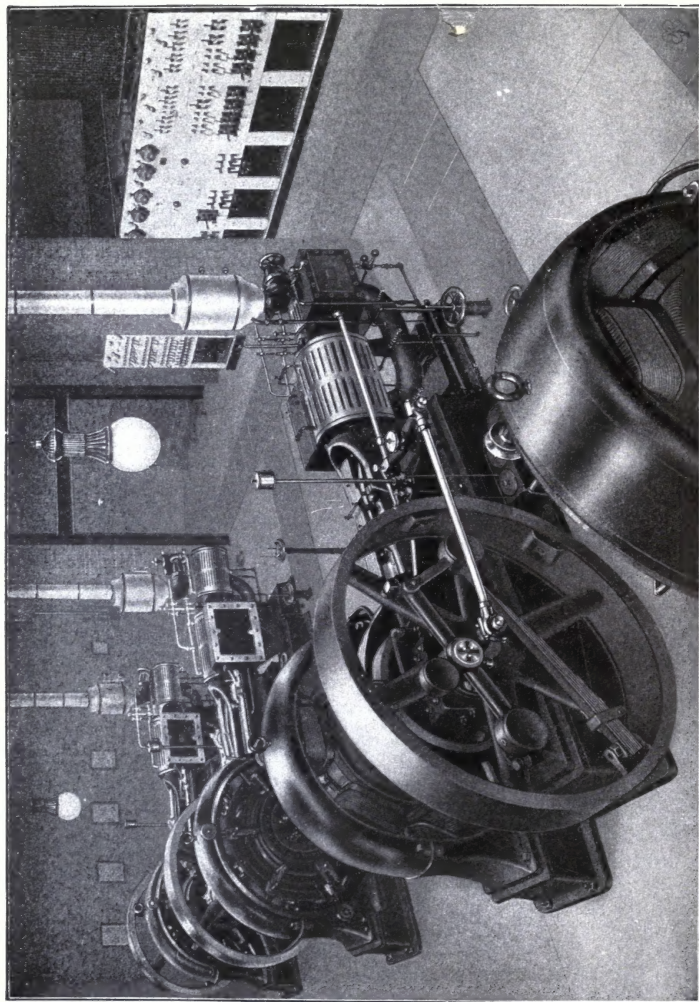
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Cyclopedia *of* Engineering

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AUTOMOBILES, MARINE AND LOCOMOTIVE WORK, HEATING AND
VENTILATING, COMPRESSED AIR, REFRIGERATION, DY-
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Grateful acknowledgment is here made also for the invaluable co-operation of the foremost engineering firms, in making these volumes thoroughly representative of the best and latest practice in the design and construction of steam and electrical machines; also for the valuable drawings and data, suggestions, criticisms, and other courtesies.

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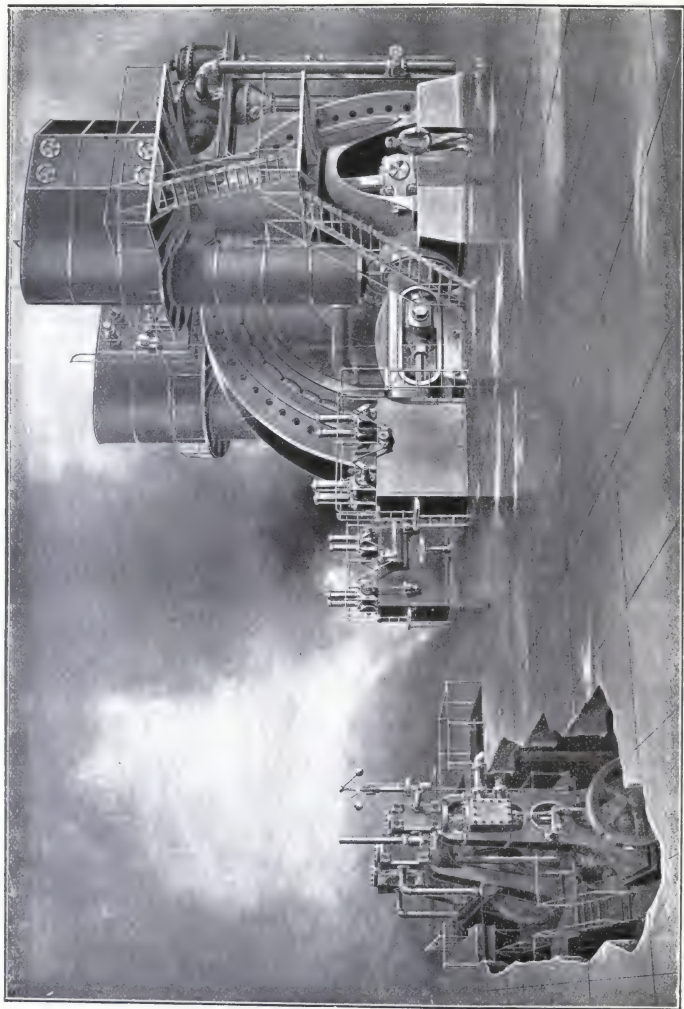
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Foreword

THE rapid advances made in recent years in all lines of engineering, as seen in the evolution of improved types of machinery, new mechanical processes and methods, and even new materials of workmanship, have created a distinct necessity for an authoritative work of general reference embodying the accumulated results of modern experience and the latest approved practice. The Cyclopedia of Engineering is designed to fill this acknowledged need.

¶ The aim of the publishers has been to create a work which, while adequate to meet all demands of the technically trained expert, will appeal equally to the self-taught practical man, who may have been denied the advantages of training at a resident technical school. The Cyclopedia not only covers the fundamentals that underlie all engineering, but places the reader in direct contact with the experience of teachers fresh from practical work, thus putting him abreast of the latest progress and furnishing him that adjustment to advanced modern needs and conditions which is a necessity even to the technical graduate.

¶ The Cyclopedia of Engineering is based upon the method which the American School of Correspondence has developed and successfully used for many years in teaching the principles and practice of Engineering in its different branches.

¶ The success which the American School of Correspondence has attained as a factor in the machinery of modern technical and scientific education is in itself the best possible guarantee

for the present work. Therefore, while these volumes are a marked innovation in technical literature—representing, as they do, the best ideas and methods of a large number of different authors, each an acknowledged authority in his work—they are by no means an experiment, but are, in fact, based on what has proved itself to be the most successful method yet devised for the education of the busy man. The formulæ of the higher mathematics have been avoided as far as possible, and every care exercised to elucidate the text by abundant and appropriate illustrations.

¶ Numerous examples for practice are inserted at intervals; these, with the text questions, help the reader to fix in mind the essential points, thus combining the advantages of a textbook with those of a reference work.

¶ The Cyclopedia has been compiled with the idea of making it a work thoroughly technical yet easily comprehended by the man who has but little time in which to acquaint himself with the fundamental branches of practical engineering. If, therefore, it should benefit any of the large number of workers who need, yet lack, technical training, the publishers will feel that its mission has been accomplished.

¶ Grateful acknowledgment is due the corps of authors and collaborators—engineers and designers of wide practical experience, and teachers of well-recognized ability—without whose co-operation this work would have been impossible.



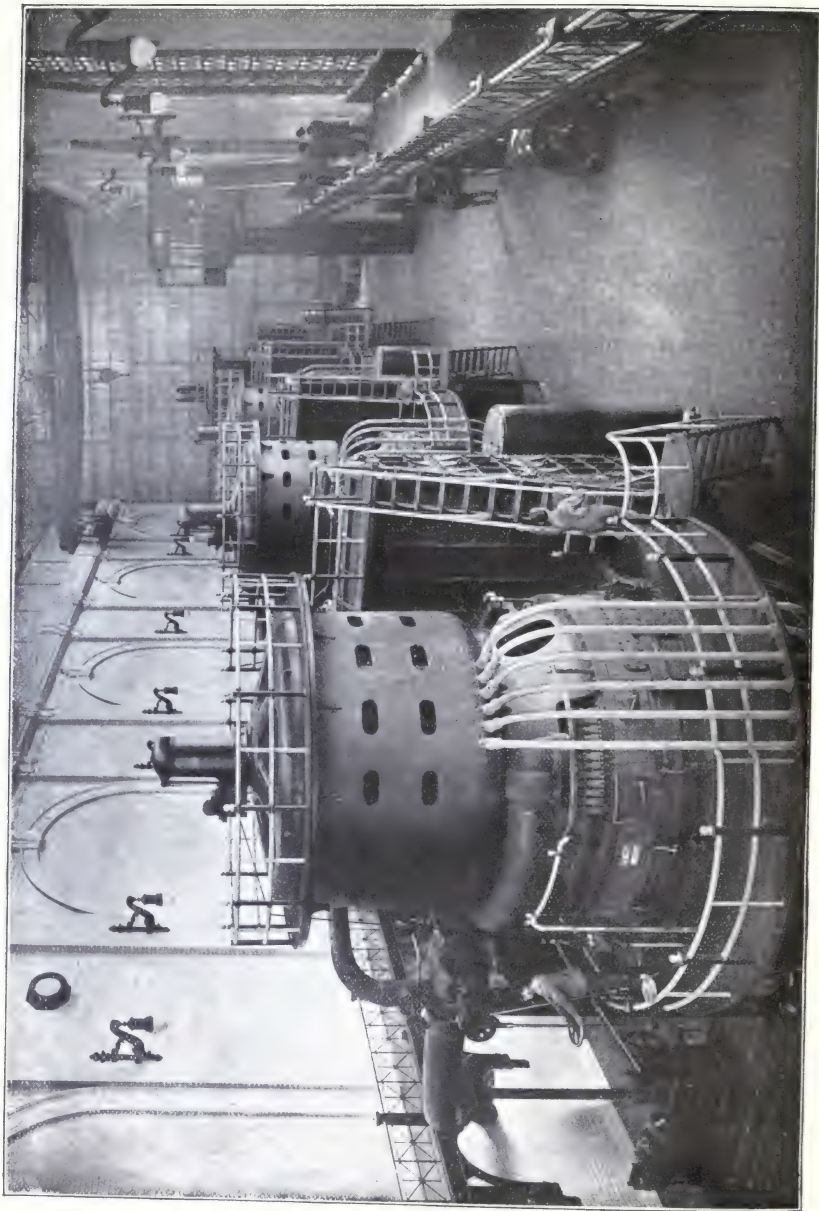
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SOME OF THE GENERATORS IN THE WORLD'S LARGEST CENTRAL STATION

View taken in Flisk Street Station of the Commonwealth Electric Company, Chicago, Ill. The total capacity of the station is 176,000 kw. Of the four im-

HEAT.

Until the time of Count Rumford and Sir Humphry Davy the most widely accepted notion of the nature of heat was that it was an elastic fluid, penetrating the pores of all matter and filling the spaces between molecules. To this fluid the name caloric was given. In order to explain the various manifestations of heat, it was necessary to ascribe various properties to the caloric fluid, such as indestructibility, absence of weight, differences in the intensity of its affinity for different kinds of matter, and so on. But these explanations were not borne out by experiment; and in 1798 Count Rumford showed, by a series of experiments with a blunt boring bar and a brass cannon, that the heat developed in boring the cannon had no relation to anything but the friction of the apparatus. He thus proved that heat could not be a material substance, for it was produced without limit from a limited quantity of matter, provided only that the motion were maintained; and he announced his conviction that heat was in reality a form of motion.

Sir Humphry Davy showed that two pieces of ice might be melted by rubbing them together. He thus proved that heat can be produced by the expenditure of work only. This leads to the important conclusion that heat must be a form of energy, since it can be produced from energy, and (as we know in the case of the steam engine) can itself be converted into well-recognized forms of energy.

Heat-energy is now understood to lie in the rapid irregular vibrations of the molecules of which all matter is composed. We shall see that the idea of rapidly-moving molecules affords a ready explanation of many of the phenomena of heat.

This motion may be communicated from one body to another; or, in other words, heat may flow from one body to another. The condition that determines which way the flow will take place is called **temperature**.

Temperature, then, has nothing to do with quantity of heat. If a spoonful of water be dipped from a full pail, it is clear that the quantities of heat in the two masses of water are very unequal, yet there is no tendency for heat to travel from either to the other.

Our study of the phenomena and laws of heat thus naturally divides itself as follows:

1. The measurement of temperature, or thermometry.
2. The measurement of quantities of heat, or calorimetry.
3. The relation between heat and mechanical work, or thermodynamics.

Equality of temperature may be estimated quite accurately simply by touching two bodies with the hand, provided they are of a similar nature and neither very hot nor very cold. The power which enables us to do this is called the temperature sense. It does not help us much, however, if the bodies are of very different nature, nor does it tell us whether they are actually hot or cold. The sensation really depends on the rate at which heat is communicated to (or taken from) the hand, and this depends on the temperature of the hand as well as on the nature of the material tested.

A simple experiment will illustrate this. Place the right hand in ice-water and the left hand in hot water; after a minute withdraw them and place them simultaneously in water just drawn from the faucet. It will seem warm to the right hand and cold to the left hand, because in the first case heat passes from the water to the hand, and in the second case from the hand to the water. Again, a stone in winter feels colder to the hand than a piece of fur or woolen cloth. The stone conducts the heat away from the hand faster than the fur does, and thus gives the sensation of a lower temperature.

THERMOMETERS.

Instruments for the measurement of temperature are called **thermometers**. In designing a thermometer we may use any substance one of whose properties varies continuously with the temperature. Among the properties most convenient for use are:

1. Expansion; used for ordinary temperatures.
2. Change of electrical resistance; used for very low temperatures.
3. Thermo-electric effects; used for very high temperatures.

The first of these is discussed in this paper; the principles on which the others depend will be explained in the Instruction Papers on Electricity.

Besides these, many other properties of substances that depend on temperature are useful in special cases. For example, when a piece of polished steel is heated its surface changes color, each color corresponding to a certain definite temperature. The process of **tempering** consists in heating the previously hardened articles until they assume the proper temperature, as shown by their color, and then plunging them again into cold water or oil. In this way each piece is made to indicate its own temperature without possibility of mistake.

Liquid Thermometers. In the most common form of thermometer, temperature is measured by the expansion of mercury in glass. On the end of a glass tube of very fine bore, a bulb is blown (see Fig. 1), and the bulb and part of the tube are filled with mercury. The whole is then heated until the mercury completely fills the tube, after which it is sealed and allowed to cool. The space in the tube above the mercury is thus entirely freed from air. Changes in temperature cause the mercury to expand or contract, and the liquid in the tube will rise or fall accordingly.

But the thermometer thus made is not yet ready for use. It must have its divisions properly spaced and in the right places on the tube. All thermometers for accurate work should have their scales engraved on the tube itself, and not on a plate attached to it. Before we can engrave the scale we must know at least two points on the stem that correspond to known temperatures. The two points commonly taken are known as the **freezing point** and the **boiling point**.

The freezing point can easily be found by putting the thermometer into a mixture of clean pounded ice (or snow) and water. The boiling point is found by immersing the whole thermometer in steam from boiling water. The freezing point is always the same under ordinary conditions, but the temperature of the boiling point rises or falls slightly as atmospheric pressure increases or decreases.



Fig. 1.

Thermometer Scales. Two scales of temperature are in use. On the *Fahrenheit* scale, devised about 1714, the boiling point is marked 212° , and the freezing point 32° , there being thus 180 degrees between them. The *Centigrade* scale, which is more convenient for scientific work, has its boiling point marked 100° and its freezing point marked 0° .

We may convert Centigrade to Fahrenheit temperatures in the following way :

Since 100 Centigrade degrees cover the same temperature interval as 180 Fahrenheit degrees, one Centigrade degree is $\frac{180}{100}$ or $\frac{9}{5}$ as long as one Fahrenheit degree. Hence a temperature of m degrees Centigrade is equal to $\frac{9}{5}m$ Fahrenheit degrees above the Centigrade zero. But this point is marked 32° on the Fahrenheit scale, consequently the total reading on the Fahrenheit thermometer will be

$$\frac{9}{5}m + 32.$$

The formula for changing a temperature C° Centigrade to its Fahrenheit equivalent F° , therefore, is

$$F^{\circ} = \frac{9}{5}C^{\circ} + 32;$$

and by transposing we obtain the corresponding formula,

$$C^{\circ} = \frac{5}{9}(F - 32).$$

EXAMPLES FOR PRACTICE.

1. To what temperature F does $58^{\circ}C$ correspond?

Ans. $136.4^{\circ}F$.

2. To what temperature C does $149^{\circ}F$ correspond?

Ans. $65^{\circ}C$.

3. The difference between the temperatures of two bodies is $45^{\circ}F$. What is it in Centigrade degrees?

Ans. $25^{\circ}C$.

4. Lead melts at $327^{\circ}C$. What is its melting point on the Fahrenheit scale?

Ans. $620^{\circ}F$.

Temperatures below the zero point can be dealt with by calling them negative and using them with a minus sign.

Example. To what temperature F does -20°C correspond?

$$\begin{aligned}\text{Solution. } F^{\circ} &= \left(\frac{9}{5} \text{ of } -20 \right) + 32 \\ &= -36 + 32 = -4^{\circ}. \quad \text{Ans.}\end{aligned}$$

EXAMPLES FOR PRACTICE.

1. To what temperature F does -18°C correspond?

Ans. -0.4°F .

2. To what temperature C does -40°F correspond?

Ans. -40°C .

The temperature T_s of steam in Centigrade degrees is given by the following formula:

$$T_s = 100^{\circ} + \frac{3}{80}(H - 760),$$

where H is the barometric pressure in millimeters.

In Fahrenheit degrees the temperature T'_s is

$$T'_s = 212^{\circ} + 1.71(H'' - 29.92),$$

where H'' is the barometric pressure in inches.

When the barometer stands at exactly 760 millimeters or 29.92 inches, the temperature of steam is therefore 100°C or 212°F . The Centigrade scale is used in almost all scientific work, while the Fahrenheit scale is more common in daily life.

EXAMPLES FOR PRACTICE.

1. What is the temperature of steam when the barometer reads 772.8 millimeters?

Ans. 100.48°C .

2. What is the temperature of steam when the barometer on a mountain stands at 27.44 inches?

Ans. 207.76°F .

EXPANSION OF SOLIDS.

When the temperature of a body rises, as a rule we find an increase in its dimensions. This is called **expansion**. It depends

on the rise of temperature and on the nature of the body itself. A rod whose length is unity at 0°C will have at any other temperature t the length

$$1 + at,$$

where a is a small constant called the **coefficient of linear expansion**. If $t = 1^{\circ}$, then the length at 1° would be simply

$$1 + a \times 1 = 1 + a,$$

and the increase in length would be

$$(1 + a) - 1 = a.$$

We may therefore define the coefficient of linear expansion as the increase in length per Centigrade degree of a rod whose length is unity at 0°C . It varies a little at different temperatures and is usually larger at higher temperatures.

The following table gives the average value of the coefficient of linear expansion for various solids, between 0° and 100°C (32° to 212°F). Different specimens of the same substance sometimes give different results, and the figures do not hold for temperatures much beyond the given limits. They may, however, be used for all ordinary purposes.

COEFFICIENTS OF LINEAR EXPANSION.

	PER DEGREE CENTIGRADE.	PER DEGREE FAHRENHEIT.
Porcelain	0.00000806	0.00000448
Gas carbon	0.0000055	0.0000031
Glass	0.0000057 to 0.0000883	0.0000032 to 0.0000491
Pine wood, along grain	0.00000608	0.00000338
Cast iron	0.00001075	0.000005972
Platinum	0.00000907	0.000005039
Steel	0.00001088 to 0.00001098	0.000006044 to 0.00000610
Wrought iron	0.00001228	0.000006822
Copper	0.00001666 to 0.00001718	0.00000925 to 0.00000954
Brass	0.00001840 to 0.00001906	0.00001022 to 0.00001059
Silver	0.00001943	0.00001079
Zinc	0.00002976	0.00001653
Ice	(-12° to 0°), 0.0001050	(10° to 32°), 0.0000583

The coefficient of surface expansion may be found by multiplying the above figures by two; and the coefficient of cubical or volume expansion, by multiplying by three.

It is clear from the table that different substances expand very unequally; zinc, for example, expanding over three times as much as platinum and more than twice as much as iron. If then



Fig. 2.

we make a bar like A of Fig. 2, by riveting together a strip of zinc and one of iron, and heat it, the bar will not only lengthen but become curved, the zinc being on the convex side. If cooled below its original temperature, the bar curves the opposite way; and by fixing one end of the bar the other end may be made to show a considerable motion for small changes of temperature.

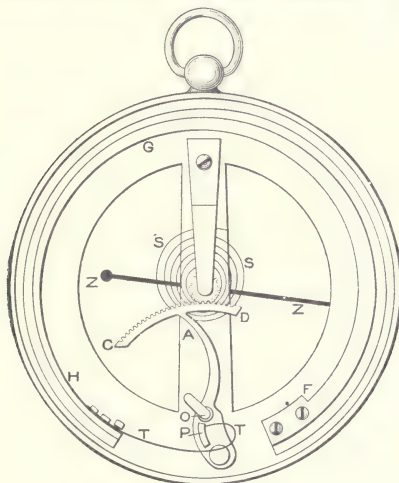


Fig. 3.

This principle is employed in some forms of metallic thermometer (see Fig. 3). The metal spring FGH is fastened at F, the remaining part being free to expand or contract. To this spring, at H, is fastened a finer spring TT, clamped at P to the arm A,

which is pivoted at O. The segment CD of a gear on the arm A operates the pinion to which the hand ZZ is attached. An additional spring SS tends to move the hand in the opposite direction. Heat causes expansion of the spring FGH, and the hand ZZ moves in a direction opposite to that of the hands of a clock. The same principle is also used in thermostats. In these instruments the free end of the bar is sometimes made to move between stops connected with electric circuits; and in this way the temperature of a furnace or a room can be easily controlled.

With the help of the table we can calculate the expansion of a rod of any length. Let l_0 represent the length of a rod at 0° C, and l_t its length at the temperature t° ; and let us find the relation between l_0 and l_t .

Since a rod of unit length at 0° will have at t° the length $1 + at$, the length of a rod l_0 times as long will be l_0 times as much, or $l_0(1 + at)$. That is,

$$l_t = l_0(1 + at). \quad (1)$$

By transposing, we may put this into the form

$$l_0 = \frac{l_t}{1 + at}. \quad (2)$$

Example. A copper wire is 65 inches long at 30° C. How long is it at 0° C?

Solution. From equation 2 we have:

$$\begin{aligned} \text{Length at } 0^\circ &= \frac{65}{1 + 0.00001666 \times 30} \\ &= \frac{65}{1.0005} = 64.967 \text{ inches. Ans.} \end{aligned}$$

Example. A sheet of zinc twenty inches by thirty is heated from 32° F to 100° F. What is its increase in area?

Solution. The surface expansion of zinc is 2×0.00001653 , or 0.00003306 per degree F.

The surface of the sheet is $20 \times 30 = 600$ sq. in.

The sheet is heated through $100^\circ - 32^\circ = 68$ degrees F.

Therefore the area of the heated sheet will be

$$600(1 + 0.00003306 \times 68) = 600(1 + 0.002248) = 601.35.$$

The increase in area is therefore 1.35 sq. in. Ans.

EXAMPLES FOR PRACTICE.

1. A brass disk has a diameter of four inches at 32°F . What is its diameter at 72°F ?

Ans. 4.00164 inches.

2. A copper tank holds ten gallons of ice water. How many gallons of boiling water will it hold?

Ans. 10.05 gallons.

Suppose the length of a rod to be given at a certain temperature t° , and we wish to find the length at some other temperature t'° . Inspection of equation 2 shows that we may write

$$l_{t'} = l_0 (1 + \alpha t').$$

We also have directly

$$l_t = l_0 (1 + \alpha t).$$

Dividing one equation by the other, we have

$$\frac{l_{t'}}{l_t} = \frac{l_0 (1 + \alpha t')}{l_0 (1 + \alpha t)};$$

or,

$$l_{t'} = l_t \frac{1 + \alpha t'}{1 + \alpha t} \quad (3)$$

A more convenient form of this equation, which will give approximately correct results, is as follows:

$$l_{t'} = l_t [1 + \alpha (t' - t)]. \quad (4)$$

Equation 4 may be used to determine the length of a bar which has been heated through a known temperature, when the original length and the coefficient of expansion are known.

EXAMPLES FOR PRACTICE.

1. A rod of copper is 10 feet long at 25°C . What will be its length at 85° ?

Ans. 10.01 feet, nearly.

2. A bar of wrought iron is 200 inches long at 40°F . What will be its length at 148°F ?

Ans. 200.147 inches.

3. If the extreme difference between summer and winter temperatures is 100 degrees F, what will be the change in length of an iron bridge which is 250 feet long in summer?

Ans. 0.1705 foot shorter.

EXPANSION OF LIQUIDS.

In the case of liquids and gases we have to deal only with cubical expansion, since fluids have no definite form. The expansion of liquids is much greater than that of solids. For mercury the average coefficient between 0°C and 100°C is 0.0001825 per degree. For other liquids the expansion increases rapidly with the temperature and is very great at high temperatures. The following table gives some values for three common liquids.

TEMP.	WATER.	ALCOHOL.	ETHER.
0°	1.	1.	1.
10°	1.0001	1.0105	1.0152
20°	1.0016	1.0213	1.0312
30°	1.0041	1.0324	1.0483
40°	1.0076	1.0440	1.0665

Water presents a partial exception to the increase of volume by rise of temperature. As its temperature rises from 0° (ice just melted) to 4°C , it contracts instead of expanding, the amount of contraction being 129 parts in a million. Above 4° it expands like any other liquid.

This curious fact is of immense importance in nature. As the water of rivers and lakes cools, it becomes denser and sinks, the coldest water thus going to the bottom until 4° is reached. Below this temperature, however, the water becomes lighter as it cools, and stays at the surface. Ice thus forms first at the surface and the life beneath is protected, as ice is a poor conductor of heat. If the water contracted down to the freezing point, ice would form from the bottom up, and a pond would become a solid mass which would probably never thaw completely.

An interesting application of the expansion of liquids is in the mercurial pendulum used in large clocks (Fig. 4). The pendulum rod carries at its end a jar of glass or iron holding a quan-

tity of mercury. A rise in temperature lengthens the rod and lowers the center of gravity of the pendulum; but the mercury also expands and rises in the jar, producing the opposite effect. In this way, by using the proper amount of mercury, it is possible to make a pendulum whose vibrations are unaffected by changes in temperature.

EXPANSION OF GASES.

If we partially fill a bladder with air and place it near a fire, it will become distended, showing that the air has expanded. This expansion is practically the same for all gases, and, for each degree, is $\frac{1}{273}$ or 0.00366 of their volume at 0°C ; for each degree F a gas will expand $\frac{5}{9}$ as much, or $\frac{1}{491}$ of its volume at 32°F . These figures assume that the pressure on the gas remains constant.

If then we have a quantity of gas V at 0°C , at 1° it will have the volume $\frac{274}{273}V$, at 2° it will have the volume $\frac{275}{273}V$, and so on. We may express the general law as follows:

If V_0 be the volume at 0°C , then the volume V_t at any other temperature t will be

$$V_t = V_0(1 + \frac{t}{273}). \quad (5)$$

Or, in decimal form,

$$V_t = V_0(1 + 0.00366t). \quad (6)$$

If t is below zero, we subtract the second term instead of adding it.

With this formula we may work exactly as with the formulas for the expansion of solids on page 11.

Example. Find the volume at 150°C of a gas measuring 10 cubic centimeters at 15° .

Solution. Applying formula 6 twice, we obtain

$$V_{150} = V_0(1 + 0.00366 \times 150) = 1.549 V_0;$$

$$V_{15} = V_0(1 + 0.00366 \times 15) = 1.0549 V_0.$$



Fig. 4.

Therefore

$$V_{150} = V_{15} \frac{1.549}{1.0549} = 10 \times \frac{1.549}{1.0549} = 14.68 \text{ cub. cm., nearly.}$$

EXAMPLES FOR PRACTICE.

1. What will be the volume of 400 cubic inches of oxygen at 0°C, when heated to 30°?

Ans. 444 cubic inches, nearly.

2. If 160 cubic centimeters of hydrogen be measured at 50° C, what will be the volume of the gas at —50°?

Ans. 110.49 cub. cm., nearly.

3. If 1,750 cubic feet of coal gas at 20°C are cooled to 0°, what will be the volume?

Ans. 1,630.6 cubic feet.

If the temperatures are expressed on the Fahrenheit scale, formula 6 becomes

$$V_{t'} = V_{32} [1 + 0.002035 (t' - 32)], \quad (7)$$

where t' is the temperature F.

EXAMPLES FOR PRACTICE.

1. 1,000 cubic feet of air are heated from 32° F to 90° F. What is the increase in volume?

Ans. 118 cub. ft.

2. 360 cubic feet of nitrogen at 70° F are cooled to 10° F. What is the volume after cooling?

Ans. 319.2 cub. ft.

When a quantity of gas confined in a given space is heated its pressure rises, and if the volume of the gas is kept constant the increase of pressure is very nearly the same as the above-described increase of volume at constant pressure. We may therefore deal with pressure changes due to temperature just as with changes of volume, employing formulas 6 and 7 as before.

EXAMPLE FOR PRACTICE.

A closed iron tank contains air at 50 lbs. pressure at 32° F. What will the pressure be if the temperature rises to 68° F, neglecting the effect of the expansion of the tank?

Ans. 53.66 lbs.

Since a reduction of temperature from 0°C to -1°C is accompanied by a loss of $\frac{1}{273}$ of the pressure of a gas, it would appear that by lowering the temperature to -273°C ($=-459^{\circ}\text{F}$) all the gas pressure would disappear, and the molecules would come completely to rest. But this means, from our definition of heat, the total absence of heat energy. This point, therefore, is called the **absolute zero of temperature**. Of course it can never be reached experimentally, but recent researches have carried the range of available temperatures far down toward it. By the evaporation of liquid and solid air and hydrogen, a temperature of -260°C has been attained.

Thus we see that to reduce ordinary temperatures to absolute temperature we add 273 if we are using Centigrade units, or 461 for Fahrenheit units.

LIQUEFACTION.

When heat is applied to an amorphous substance like glass or pitch, it changes gradually from a solid to a liquid, and there is no definite point at which melting occurs; but for most crystalline substances the change from solid to liquid is well marked. For such substances, melting (also called fusion) takes place according to the following laws:

1. Every substance melts at a certain temperature, which is always the same if the pressure on the substance is the same.
2. After fusion begins, the temperature of the mass remains at the melting point until the solid is completely melted.
3. In cooling, the substance solidifies at the temperature of melting.

TABLE OF MELTING POINTS.

	Centi- grade.		Centi- grade.
Ether,	-117°	Zinc,	418°
Mercury,	-39.4	Silver,	908
Ice,	0	Gold,	1072
Paraffin,	46	Copper,	1082
Wood's metal,	65 to 70	Cast iron,	1100 to 1200
Sulphur,	114	Wrought iron,	1600
Tin,	232	Platinum,	1775
Lead,	327	Iridium,	1950

* NOTE. Some authorities quote -461°F .

Most substances increase in volume on melting, but some contract. The reverse change takes place on solidifying. Good castings can be made only from those metals or alloys which expand on solidifying, like cast iron and type-metal. Gold, aluminum, lead and silver must be stamped to get sharp impressions.

VAPORIZATION.

Since heat is the rapid, irregular vibratory motion of the molecules, it follows that if we add heat to a body we increase this motion. At a certain stage the vibration is so vigorous that the molecules (if the body is a solid) can no longer hold fast to one another, and the solid literally falls to pieces, that is, it melts. By applying more heat to the liquid and still further raising its temperature, we may finally reach a point at which some of the molecules are moving so violently as to escape into the air, altogether free from one another's influence. We then have a **vapor**, and the change into this æriform condition is called **vaporization**.

If vaporization takes place slowly, and only at the surface of a liquid, it is called **evaporation**. Evaporation will be hastened by anything that facilitates the escape of molecules from the liquid surface, as by increasing the temperature of the liquid, lowering the pressure on it, or causing a breeze to play over the surface.

The fact that heat is due to molecular motion explains why evaporation is a cooling process. Naturally those molecules will escape first whose motion is most violent, that is, whose temperature is highest. The more sluggish (and therefore colder) molecules stay behind. Thus, as the liquid evaporates, the departing molecules take with them more than their proportionate share of heat, and the remaining liquid grows colder.

Cooling by evaporation may be illustrated by a simple experiment. Drop about a teaspoonful of water on a table or smooth board, and set a small tin dish on the water. Pour three or four tablespoonfuls of ether into the dish, and blow upon it with a pair of bellows. After two or three minutes of vigorous blowing, the dish will be found frozen fast to the board. (Caution.—*Keep ether away from lights.* Ether vapor is highly inflammable.)

When water is heated over a flame, the air (or any other gas) present is first driven off in tiny bubbles which rise to the surface and escape without noise. When the water nearest the flame is raised to the boiling point, bubbles of vapor are formed, which also rise through the water, but are condensed by the cooler layers before getting to the surface. This formation and condensation of steam bubbles produces the sound known as singing or simmering. The "water hammer" in steam pipes is of a somewhat similar nature but on a larger scale. When the entire mass is heated to the boiling point, the steam bubbles rise to the surface and break, discharging their contents into the air with a characteristic noise. This stage is called **ebullition** or boiling.

Like air, steam is colorless, transparent and invisible. What is commonly called "a cloud of steam" is really a cloud of fine water particles condensed from steam. Observe any steam jet, and notice that at the end of the pipe nothing whatever can be seen, the jet becoming visible only after it has gone far enough from the pipe to be cooled and condensed.

The increase of volume by vaporization is usually very great. For example, a cubic inch of water will make 1,661 cubic inches of steam at atmospheric pressure.

By increasing the pressure on the surface of a boiling liquid, we make it more difficult for the molecules to escape; they cannot escape unless given more motion, that is, unless they have a higher temperature than before. In other words, an increase of pressure raises the boiling point. The following table gives the boiling point of water under different pressures, as measured by a steam gage :

BOILING POINT OF WATER.

GAGE PRESSURE.	TEMPERATURE, FAHR.
0 (atmosphere)	212°
50 lbs.	297.4
100 "	337.6
150 "	365.7
200 "	387.8

The laws of vaporization are similar to the laws of fusion given on page 15. The following table gives the boiling points in

degrees Centigrade of some liquids, under a pressure of one atmosphere :

BOILING POINTS.

Liquid air,	— 188°	Chloroform,	61.2°
Ammonia,	— 38.5	Alcohol,	78.4
Sulphurous anhydride,	— 10.1	Mercury,	357.
Ether,	34.9	Sulphur,	444.5

DISTILLATION.

The difference in the boiling points of substances has an important application in the arts, in the separation of liquids from

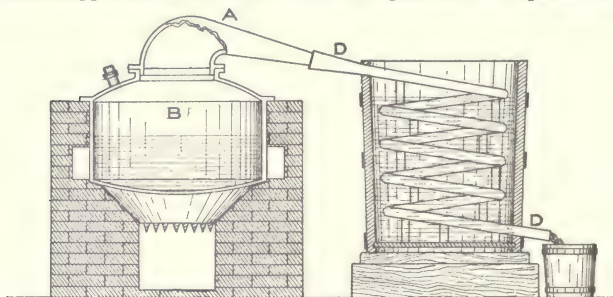
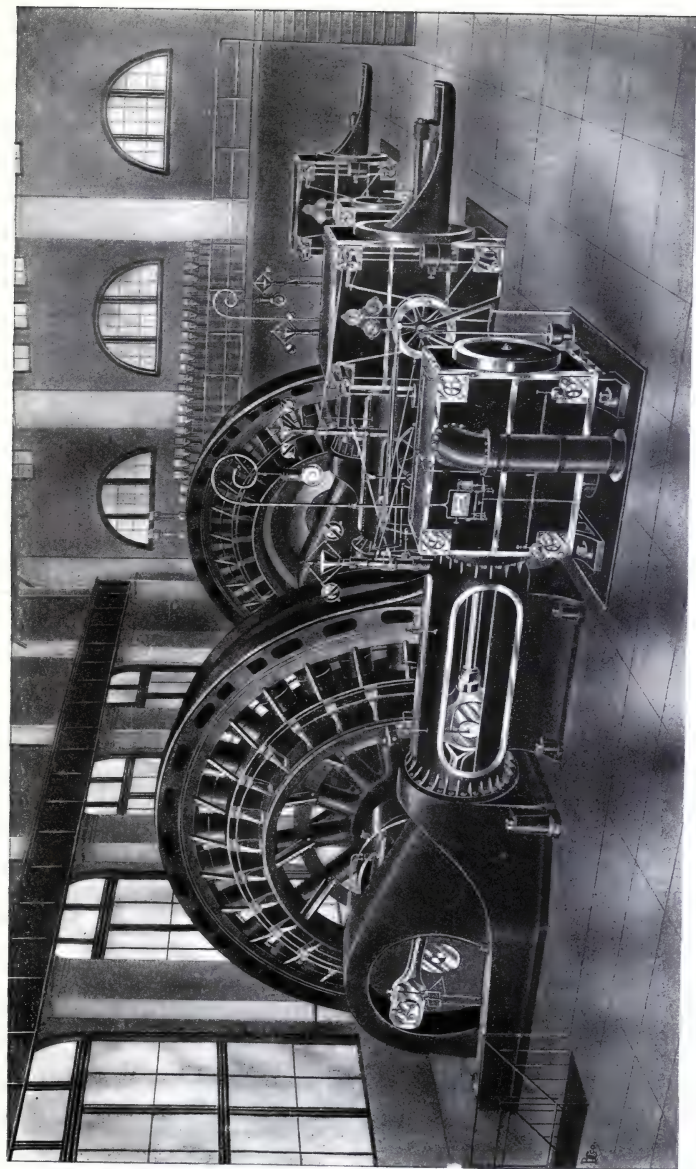


Fig. 5.

solids, or of liquids from each other. The simple removal of a liquid from a solid, as in evaporating brine to recover the salt, needs no special appliances ; but when the evaporated liquid is to be saved, an apparatus called a **still** is used, and the process is called **distillation**.

A still consists essentially of two parts: a *retort* in which the liquid is vaporized, and a *condenser* in which it is reduced to liquid again. Fig. 5 shows a form of the apparatus for separating a liquid from a solid, or one liquid from another of different boiling point, such as alcohol and water. The mixture is poured into the retort B, and then heated to about 90°C, which is above the boiling point of alcohol but below that of water. The vaporized alcohol escapes through A to the *worm* D. This is a simple



FULTON CORLISS HEAVY-DUTY CROSS-COMPOUND CONDENSING ENGINES, DIRECT-CONNECTED TO 2,250-K. W. RAILWAY GENERATORS
Fulton Iron Works, St. Louis, Mo.



helical coil of pipe surrounded by cold water, and serving to condense the vapor, which runs out as a liquid at the bottom. The cooling water is constantly changed by supplying fresh cold water at the bottom and drawing off the heated water from the top.

This process is called **fractional distillation**, and is carried out on an enormous scale in the refining of petroleum. When the distillate is to be very pure, it is necessary to repeat the operation one or more times. When practicable, especially with inflammable liquids, the heating is done by steam pipes supplied from a distant boiler.

THE MEASUREMENT OF HEAT. Heat Units.

There are two units of measurement for determining quantities of heat. The **British thermal unit** (often abbreviated B. T. U.) is the amount of heat required to raise one pound of water from 59° to 60° Fahrenheit. The French unit, or **calorie**, is the amount of heat required to raise the temperature of one gram of water from 15° to 16° Centigrade. The former is much used in engineering calculations involving steam and fuels, and the latter in all other scientific work.

LATENT HEAT.

If we put a block of very cold ice into a vessel over a flame and insert a thermometer into the ice, we shall observe the thermometer rise to 0° C, at which point the ice begins to melt. The temperature of the ice and water then shows no further change until all the ice has melted, though the heat is applied continuously. Only after the melting is complete will the temperature of the water begin to rise. It will then increase until 100° C is reached, when ebullition begins, the temperature not rising above 100° until all the water has boiled away.

We thus see that in changing from ice to water and from water to steam there is absorbed a considerable quantity of heat which does not show on the thermometer. The quantities of heat absorbed in the processes of fusion and vaporization are called the *latent heat of fusion* and the *latent heat of vaporization* respectively.

The following example shows how the latent heat of fusion of ice may be measured. If we mix a gram of water at 80°C with a gram at 0° , we get, as we should expect, two grams at 40° . But if we mix a gram of water at 80° with a gram of ice at 0° , we get two grams of water as before, but the temperature is 0° instead of 40° . The heat which in the first case raised the temperature of the water has in the second case been needed merely to melt the ice. The calculation of the latent heat is made in the following way:

One gram of water falling through 80 degrees of temperature will give out 1×80 , or 80 calories. This quantity of heat is required to change one gram of ice at 0° into water at 0° . Therefore the latent heat of fusion of ice is 80; in other words, the heat which will just melt a quantity of ice will raise 80 times as much water one degree C.

By a somewhat similar method it is found that the latent heat of vaporization of water at atmospheric pressure is 536.5. That is, to evaporate one gram of water (already at the boiling point) will require as much heat as would raise the temperature of 536.5 grams one degree, or 5.365 grams from freezing to boiling (0° to 100°C).

Expressed in terms of the Fahrenheit degree and the British thermal unit, the latent heats of fusion and vaporization are 144 and 966 respectively.

The large values of these quantities are of the greatest importance both in nature and in the arts. The great amount of heat necessary to melt the ice of winter makes the melting a slow process, and lessens the danger of destructive floods in the spring. In the autumn the water in freezing gives out again the heat absorbed in melting, and the transition to winter is thus rendered less abrupt.

Since a pound of steam in condensing will give out as much heat as 53.65 pounds of water cooling from 100°C to 90°C , or from 90° to 80° , it follows that steam pipes for heating may be made smaller than water pipes for the same service. It also shows the value of steam as a carrier of heat; and in the arts advantage of this is taken in innumerable ways. (See also page 19).

SPECIFIC HEAT.

When equal quantities of different substances are raised equally in temperature, different amounts of heat are required; and in cooling through equal temperature intervals different substances give out different amounts of heat.

For example, if we mix a pound of water at 80°C with a pound at 0°C , we get two pounds at 40° ; but if we pour a pound of lead shot at 80° into a pound of water at 0° , the resulting temperature will be only 2.3° . A pound of lead, therefore, falling through 77.7 degrees of temperature, is able to raise a pound of water only 2.3 degrees. The fall of temperature of the hot body is nearly twice as great as in the first case, and the heat given out in the fall only about one-seventeenth as much. The heat capacity of the lead is therefore much less than that of the water.

If we know how much heat will raise the temperature of a given substance a certain amount, and how much is required to raise the temperature of an equal quantity of water by the same amount, then the ratio of these two quantities is called the specific heat of the substance. In other words, if we take the specific heat of water as our standard (as we practically do in defining the units of heat), the specific heat of a substance is expressed by the number of heat units required to raise the unit quantity of the substance one degree in temperature.

One of the simplest methods of determining specific heat is by mixing the substance with water. Suppose that 6 pounds of mercury at 100°C are poured into 2 pounds of water at 0°C , and that the resulting temperature of the "mixture" is 9° . The specific heat S of the mercury can then be found as follows:

In falling from 100° to 9° the 6 pounds of mercury give out $6 \times (100 - 9) \times S$, or $546 S$ heat units. These have gone to heat 2 pounds of water from 0° to 9° , which requires 2×9 , or 18 heat units. Hence we may write

$$546 S = 18$$

Therefore

$$S = 0.033.$$

EXAMPLE FOR PRACTICE.

Half a pound of a metal at 212°F is dropped into one pound of water at 68°F . The temperature of the mixture is then observed to be 76°F . What is the specific heat of the metal?

Ans. 0.117.

TABLE OF SPECIFIC HEATS.

Hydrogen,	3.4090	Wrought iron,	0.1124
Alcohol,	0.602		
Ammonia (gas),	0.5084	Copper,	.0949
Ice,	.5040		
Air,	.2375	Zinc,	.0935
Aluminum,	.2122	Tin,	.0562
Glass,	.193 to .198	Mercury,	.0330
Cast iron,	.1298	Lead,	.0314
Steel,	.1181		

With the foregoing principles and the help of suitable tables, many problems can be solved. For example, let us find how many calories will be required to convert 10 grams of ice at -12°C into steam at 100°C .

Solution. Required to raise the ice from -12° to 0° ,

$$10 \times 12 \times 0.504 = 60.48 \text{ calories.}$$

Required to melt the ice,

$$10 \times 80 = 800 \text{ calories.}$$

Required to raise the water from 0° to 100° ,

$$10 \times 100 = 1,000 \text{ calories.}$$

Required to vaporize the water,

$$10 \times 536.5 = 5,365 \text{ calories.}$$

Total number of calories required,

$$60.48 + 800 + 1,000 + 5,365 = 7,225.5 \text{ (nearly).}$$

EXAMPLES FOR PRACTICE.

1. What weight of water at 75°C will just melt 15 pounds of ice at 0° ?

Ans. 16 pounds.

2. One kilogram of water at 40° , 2 kilograms at 30° , 3 kilograms at 20° , and 4 kilograms at 10° are mixed. Find the temperature of the mixture.

Ans. 20° .

3. How many heat units will be required to melt 5 grams of ice at -20°C ? How many grams of water at 50°C would do it?

Ans. 450.4 heat units; 9.01 grams.

If we wish to use Fahrenheit degrees and British thermal units in our calculations, it is necessary to remember that the numbers representing the heats of fusion and of vaporization are different, but that the specific heat, which is a mere ratio, is the same in both systems.

For example, let us find how many B. T. U. are required to convert 12 lbs. of ice at 10°F into steam at 212°F .

Solution. Required to raise the ice from 10°F to 32°F ,

$$12 \times 22 \times 0.504 = 133.056 \text{ B. T. U.}$$

Required to melt the ice,

$$12 \times 144 = 1,728 \text{ B. T. U.}$$

Required to raise the water from 32° to 212° ,

$$12 \times 180 = 2,160 \text{ B. T. U.}$$

Required to vaporize the water,

$$12 \times 966 = 11,592 \text{ B. T. U.}$$

Total number of B. T. U. required,

$$133.056 + 1,728 + 2,160 + 11,592 = 15613 \text{ (approx.)}$$

EXAMPLE FOR PRACTICE.

How many B. T. U. are required to convert 10 lbs. of ice at 15°F into steam at 212°F ?

Ans. 12,985 B. T. U.

For ordinary purposes we may proceed as above; but as the specific heat and latent heat of water vary for different temperatures, we must, where great accuracy is necessary, employ a table of the properties of steam and water.

THE PROPERTIES OF STEAM.

The relation between the external pressure and the boiling point of water is a perfectly definite one, but cannot be exactly expressed by any mathematical equation. In dealing with this and other properties of steam and water, it is therefore customary to refer to suitable tables where the values are given, as determined by experiment. Such tables are called **steam tables**, and are much used in engineering calculations.

In the following table are given (1) the pressure above absolute vacuum, (2) the corresponding temperature, (3) the amount of heat in B. T. U. required to raise a pound of water from 32°F to the given temperature, (4) the amount of heat in B. T. U. required to vaporize a pound of water at the given temperature; (5) equals the sum of (3) and (4).

A steam gage measures pressures above the atmospheric pressure; hence, when readings are taken from a steam gage, the barometric pressure (averaging 14.7 lbs. per sq. in., or in round numbers 15 lbs.) must be added to obtain the "absolute" pressure.

With a steam table we can extend considerably the range of problems like those on page 23. For example, let us find how many pounds of steam at 65 lbs. gage pressure will be needed to raise the temperature of 60 pounds of water from 50°F to 100°F .

Solution. To raise one pound of water from 50° to 100° requires 50 B. T. U.; and for 60 pounds we need 50×60 , or 3,000 B. T. U. At 65 lbs. gage pressure (80 lbs. absolute) the total heat of one pound of steam is 1,177 B. T. U., and this amount would all be available if we cooled it down to 32°F . But since the cooling is not carried below 100°F , we cannot use $100-32$,

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure in pounds persq.in. above vacuum.	Temperature in degrees Fahrenheit.	Heat in liquid 32° in units.	Heat of vaporization, or latent heat in heat units	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of 1 pound in cubic feet.	Total pressure above vacuum.
1	101.99	70.0	1043.0	1113.1	0.00299	334.5	1
2	126.27	94.4	1026.1	1120.5	0.00576	173.6	2
3	141.62	109.8	1015.3	1125.1	0.00844	118.5	3
4	153.09	121.4	1007.2	1128.6	0.01107	90.31	4
5	162.34	130.7	1000.8	1131.5	0.01366	73.21	5
6	170.14	138.6	995.2	1133.8	0.01622	61.67	6
7	176.90	145.4	990.5	1135.9	0.01874	53.37	7
8	182.92	151.5	986.2	1137.7	0.02125	47.09	8
9	188.33	156.9	982.5	1139.4	0.02374	42.12	9
10	193.25	161.9	979.0	1140.9	0.02621	38.15	10
14.7	212.00	180.9	965.7	1146.6	0.03794	26.36	14.7
15	213.03	181.8	965.1	1146.9	0.03826	26.14	15
20	227.95	196.9	954.6	1151.5	0.05023	19.91	20
25	240.04	209.1	946.0	1155.1	0.06199	16.13	25
30	250.27	219.4	938.9	1158.3	0.07360	13.59	30
35	259.19	228.4	932.6	1161.0	0.08508	11.75	35
40	267.13	236.4	927.0	1163.4	0.09644	10.37	40
45	274.29	243.6	922.0	1165.6	0.1077	9.287	45
50	280.85	250.2	917.4	1167.6	0.1188	8.414	50
55	286.89	256.3	913.1	1169.4	0.1299	7.696	55
60	292.51	261.9	909.3	1171.2	0.1409	7.097	60
65	297.77	267.2	905.5	1172.7	0.1519	6.583	65
70	302.71	272.2	902.1	1174.3	0.1628	6.143	70
75	307.38	276.9	898.8	1175.7	0.1736	5.762	75
80	311.80	281.4	895.6	1177.0	0.1843	5.426	80
85	316.02	285.8	892.5	1178.3	0.1951	5.126	85
90	320.04	290.0	889.6	1179.6	0.2058	4.859	90
95	323.89	294.0	886.7	1180.7	0.2165	4.619	95
100	327.58	297.9	884.0	1181.9	0.2271	4.403	100
105	331.13	301.6	881.3	1182.9	0.2378	4.205	105
110	334.56	305.2	878.8	1184.0	0.2484	4.026	110
115	337.86	308.7	876.3	1185.0	0.2589	3.862	115
120	341.05	312.0	874.0	1186.0	0.2695	3.711	120
125	344.13	315.2	871.7	1186.9	0.2800	3.571	125
130	347.12	318.4	869.4	1187.8	0.2904	3.444	130
140	352.85	324.4	865.1	1189.5	0.3113	3.212	140
150	358.26	330.0	861.2	1191.2	0.3321	3.011	150
160	363.40	335.4	857.4	1192.8	0.3530	2.833	160
170	368.29	340.5	853.8	1194.3	0.3737	2.676	170
180	372.97	345.4	850.3	1195.7	0.3945	2.535	180
190	377.44	350.1	847.0	1197.1	0.4153	2.408	190
200	381.73	354.6	843.8	1198.4	0.4359	2.294	200
225	391.79	365.1	836.3	1201.4	0.4876	2.051	225
250	400.99	374.7	829.5	1204.2	0.5393	1.854	250
275	409.50	383.6	823.2	1206.8	0.5913	1.691	275
300	417.42	391.9	817.4	1209.3	0.644	1.553	300
325	424.82	399.6	811.9	1211.5	0.696	1.437	325
350	431.60	406.9	806.8	1213.7	0.748	1.337	350
375	438.40	414.2	801.5	1215.7	0.800	1.250	375
400	445.15	421.4	796.3	1217.7	0.853	1.172	400
500	466.57	444.3	779.9	1224.2	1.065	.939	500

or 68 B. T. U., and the amount available is therefore $1,177 - 6 = 1,109$ B. T. U.

The quantity of steam therefore needed is

$$\frac{3,000}{1,109} = 2.705 + \text{pounds.}$$

Ans. $2.705 +$ pounds.

The small quantity of steam in this example well illustrates the great heating power of steam.

EXAMPLES FOR PRACTICE.

1. How many pounds of steam at 100 lbs. absolute pressure will raise 250 pounds of water from 50° F to 150° F?

Ans. 23.5 pounds.

2. How many pounds of steam at 35 pounds gage pressure will just melt 1,000 pounds of snow at 32° F?

Ans. 123.3 pounds.

It will be seen that the values of the total heat column in the table increase but slowly, while the pressure and temperature increase rapidly. A pound of high-pressure steam thus contains but little more heat than a pound of low-pressure steam, and consequently requires but little more fuel to produce it. Since high-pressure steam is more effective in steam engines, there is therefore a decided thermodynamic advantage in using steam of the highest practicable pressure.

Superheated Steam. From the table we see that there is a perfectly definite temperature for steam at any given pressure. Steam or other vapors in this condition are said to be **saturated**, for if the temperature is lowered some of the vapor will condense immediately into liquid. If, however, we pass steam through a separately heated pipe or chamber it is easily possible to raise its temperature by any desired amount above the given values. Steam in this condition is called **superheated**. For steam engines it has certain advantages over saturated steam, which are discussed in the Instruction Papers on the Steam Engine.

TRANSFER OF HEAT.

Heat may be transferred from one body to another by conduction, convection or radiation.

Conduction. When one end of a metal bar is heated in the fire the other end gradually becomes warmer. The heated molecules communicate their motion to their immediate neighbors; and the heat thus travels along the bar, and may be removed by a cold body at the distant end. This process is called conduction. In this way the heat of a boiler furnace is communicated to the water in the boiler.

A brass pin held in a gas flame will burn the fingers almost in-

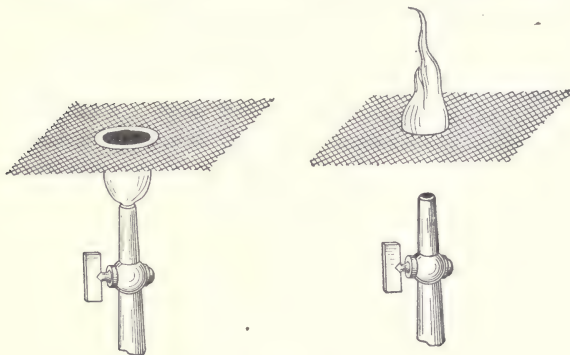


Fig. 6.

stantly, while a bit of glass may be melted at one end before the other becomes hot, and a match may be burned to the finger-tips without discomfort. It is thus clear that substances differ greatly in conductivity. There are great differences even among metals; a copper rod will conduct heat much more rapidly than an iron rod.

If a piece of wire gauze is held over an unlighted gas jet, the gas may be lighted on either side (Fig. 6), but the flame will not pass through the meshes. The wires conduct the heat away so rapidly that the gas on the other side does not get hot enough to ignite. This is the principle of the safety lamp, used in coal

mines where inflammable gases collect. The lamp flame is surrounded by a wire gauze, and thus kept from igniting the dangerous gases outside.

The following table gives the relative conductivity of several substances. The table well shows the great value of a layer of snow as a protective blanket on the earth :

RELATIVE THERMAL CONDUCTIVITIES.

Silver,	100	Lead,	7
Copper,	66 to 74	Marble,	0.0074
Brass,	18 to 23	Ice,	0.0052
Iron,	15	Snow,	0.00046

Convection. Excepting mercury, liquids and gases are poor conductors of heat ; but when such bodies are heated from below, the heated portion expands and rises through the mass, and is replaced below by a colder portion, which is heated and rises in its turn. In this way what are called **convection currents** arise, and the heat is distributed throughout the fluid by actual motion within the mass itself. The heating of houses by hot water is an application of this principle.

Convection also takes place in gases ; the winds of the atmosphere illustrate it on a large scale.

Radiation. We have seen that the molecules of a hot body are in very rapid vibration. Some of the energy of this vibration is communicated in the form of waves to the space surrounding the body. If the motion happens to lie within certain limits, the waves affect our eyes and we call them light-waves. But all such waves, visible or otherwise, represent energy which is sent out by the hot body. When they fall upon any other body, they are either reflected or absorbed and transformed into heat. Polished silver reflects over 90 per cent of the waves falling on it ; charcoal absorbs nearly all, and hence rises in temperature when exposed to the radiations of a hot body.

Energy in this form is called radiant energy. By it the heat of the sun is transmitted to the earth. Since it is in the form of ether-waves, many of the experiments ordinarily performed with

light-waves may be repeated with the radiations from a hot body, whether visible or not. The common burning-glass shows the result of bringing such rays to a focus by refraction. If a pair of concave mirrors be set facing each other, as shown in Fig. 7, and a source of heat be placed in the focus of one, a thermometer in the focus of the other will quickly show a rise in temperature though the mirrors are many feet apart.

It does not follow, however, that bodies transparent to light are equally transparent to other radiations. Glass, for example, is quite opaque to the invisible radiations that are most effective in

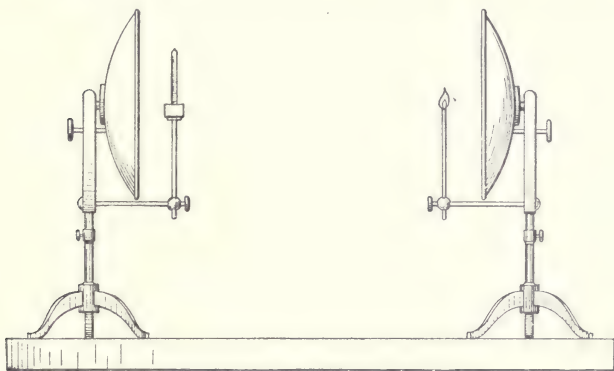


Fig. 7.

producing heat. Also, a solution of alum in water will cut off most of these radiations, while allowing the light-waves to pass freely; and a glass tank of alum water is often used in stereopticons to keep the heat of the lamp from the rest of the apparatus.

It is important to note that though the radiation from a hot body is often called radiant heat, yet in the process of transmission it is not heat at all, but a wave motion in the ether, which is energy of a very different kind. A somewhat analogous case is present in the incandescent lamp; the heat which appears at the lamp does not come along the wires as heat, but as electrical energy, which is altogether different. It is indeed transformed into heat in the lamp, but is not itself heat. In like manner,

radiant energy is transformed into heat only by falling on some body which absorbs it. It is thus possible to make a good burning-glass by shaping a piece of clear ice into the form of a lens, a thing which would clearly be impossible if the energy passing through the ice lens were in the form of heat.

THERMODYNAMICS.

This is the science which deals with the relations between heat and mechanical energy. It rests on two fundamental propositions called the first and second laws of thermodynamics.

The **first law** states that when heat is transformed into

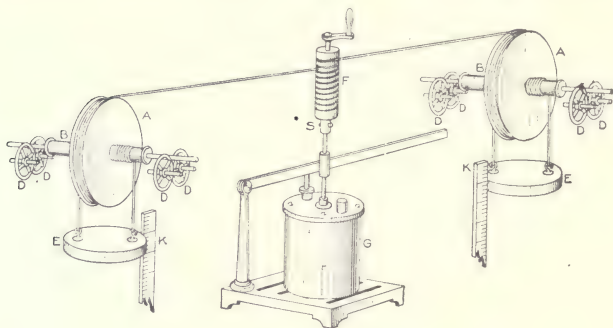


Fig. 8.

mechanical energy, or the reverse, the quantity of heat is always exactly equivalent to the quantity of mechanical energy.

The production of heat by friction is familiar to every one; but it is not so clear that there is an exact equivalence between the energy lost by friction and the heat produced. Joule was the first to establish the relation accurately. The principle of his apparatus is shown in Fig. 8. The falling weights EE turned a paddle-wheel stirrer inside the cylindrical vessel G, which was filled with water and was much like the common ice-cream freezer. The friction of the stirrer heated the water; and when the distance was measured through which the weights fell, it was possible to calculate the relation between the work done by the falling weights and the heat developed in the water.

Later experiments on a larger scale have given results which are more accurate than was possible with this apparatus. The values now accepted are the following :

427.3 kilogrammeters of work or energy are required to raise the temperature of one kilogram of water from 15° to 16° C at sea-level, in latitude 45° .

In English units, 778.8 foot-pounds of work are required to raise the temperature of one pound of water from 59° to 60° Fahrenheit at sea-level, in latitude 45° .

These values vary slightly for different places, because the weight of a pound depends on the pull of gravity, and this varies in different places ; but for most engineering purposes 779 foot-pounds would be near enough.

This law states in effect that we cannot get energy for nothing. Whenever we get work from heat, a definite quantity of heat disappears ; and whenever we convert mechanical energy into heat, we must expend 779 foot-pounds for every British thermal unit produced.

The **second law** of thermodynamics asserts that heat cannot of itself pass from a cold to a hot body. Since a hot body in cooling gives out heat which, in part at least, may be converted into work, it might seem that by cooling it indefinitely we could get an infinite amount of work from it. But the second law declares that the process stops as soon as the hot body has cooled to the temperature of its surroundings ; and if we wish to cool it further we must expend energy in so doing. It follows from this that no heat engine can convert into work all the heat which it receives. As soon as the steam (or other working fluid) has fallen to the temperature of the exhaust, the remaining heat in it is no longer available for doing useful work. If the heat is supplied at the absolute temperature T_1 and the exhaust is at the temperature T_2 , the efficiency of the engine cannot be greater than $\frac{T_1 - T_2}{T_1}$, no matter what is used as the working fluid of the engine.

THERMODYNAMICS OF PERFECT GASES.

The subject of thermodynamics cannot be fully treated by elementary methods of analysis ; in the following brief discussion, however, no advanced methods are used.

The thermodynamics of steam will be more readily understood by first taking up the simpler case of a perfect gas. Air, oxygen, nitrogen, hydrogen and some others, behave very nearly as perfect gases; others, as ammonia, carbon dioxide and sulphur dioxide, do not.

Boyle's Law. The product of the pressure P and the volume V of a perfect gas is constant if the temperature is constant; that is, if at a pressure P_1 a body of gas has the volume V_1 , and at some other pressure P_2 has the volume V_2 , then

$$P_1 V_1 = P_2 V_2 = \text{constant.}$$

Example. If 12 cubic feet of air at 135 pounds absolute pressure expand to 27 cubic feet at the same temperature, what will be the pressure? What pressure would a gage indicate?

Solution. $P_{135} V_{135} = P_{27} V_{27}$

$$135 \times 12 = P_{27} \times 27$$

Therefore $P_{27} = 60$ pounds, absolute.

Gage pressure = absolute pressure — atmospheric pressure.

$$P_{\text{gage}} = 60 - 14.7 = 45.3 \text{ pounds. Ans.}$$

EXAMPLE FOR PRACTICE.

Ten cubic feet of air at 2.3 lbs. gage are compressed until the gage pressure is 7.3 lbs. Find the volume.

Ans. 7.727 cub. ft.

Since one pound of air at 32° F occupies 12.387 cubic feet, we may calculate the product PV of a pound of air as follows:

$$V = 12.387$$

$$P = 14.7 \times 144 = 2,117 \text{ (nearly).}$$

$$\text{Hence } PV = 2,117 \times 12.387 = 26,223.$$

Law of Boyle and Charles. For a perfect gas the product PV is proportional to the absolute temperature T . In the form of an equation, this becomes $\frac{PV}{T} = \text{constant}$, or $PV = \text{constant} \times T$.

This is usually written $PV = RT$. For air we may easily calculate R as follows: We have just seen that at 32°F , $PV = 26,223$, and $T = 32 + 461$, or 493° absolute temperature. Therefore,

$$\frac{26,223}{493} = R = 53.2.$$

Example. What volume will be occupied by a pound of air at 50°F and 40 pounds pressure (absolute) per sq. in.?

Solution. $P = 40 \times 144 = 5,760$ lbs. per sq. ft.

$$T = 50 + 461 = 511^{\circ} \text{ absolute temperature.}$$

$$\text{Therefore} \quad 5,760 \times V = 53.2 \times 511.$$

$$V = \frac{53.2 \times 511}{5,760}.$$

$$V = 4.72 \text{ cub. ft. (nearly).}$$

Example. A quantity of air at 75 lbs. gage pressure and 60°F is heated to 90°F . What is the pressure?

Solution. Since the volume is unaltered, the pressure is proportional to the absolute temperature. We have

$$60^{\circ}\text{F} = 60 + 461 = 521^{\circ} \text{ absolute.}$$

$$90^{\circ}\text{F} = 90 + 461 = 551^{\circ} \text{ absolute}$$

$$\text{Therefore} \quad P_1 : P_2 :: T_1 : T_2.$$

$$P_2 = \frac{T_2}{T_1} \times P_1.$$

$$P_2 = \frac{551}{521} \times (75 + 14.7) = 94.86 \text{ lbs.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

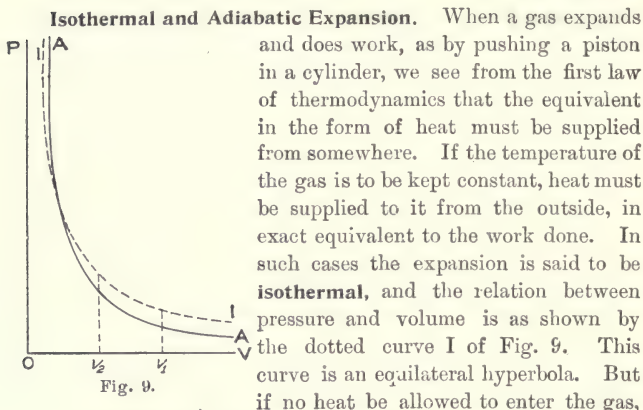
1. What is the weight of 6 cubic feet of air at 60°F and 25 pounds absolute pressure per square inch?

Ans. .78 lbs.

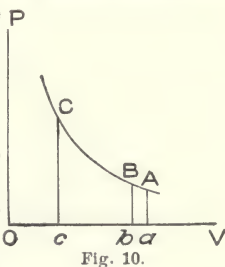
SUGGESTION. First find the volume of one pound under the given conditions.

2. A reservoir containing 4 cubic feet of air at a temperature of 40°F and a pressure of 100 pounds per square inch abs., is heated to 80°F . What will the pressure be, and how much does the air weigh?

$$\text{Ans. } \left\{ \begin{array}{l} 107.98 + \text{lbs.} \\ 2.16 + \text{lbs.} \end{array} \right.$$

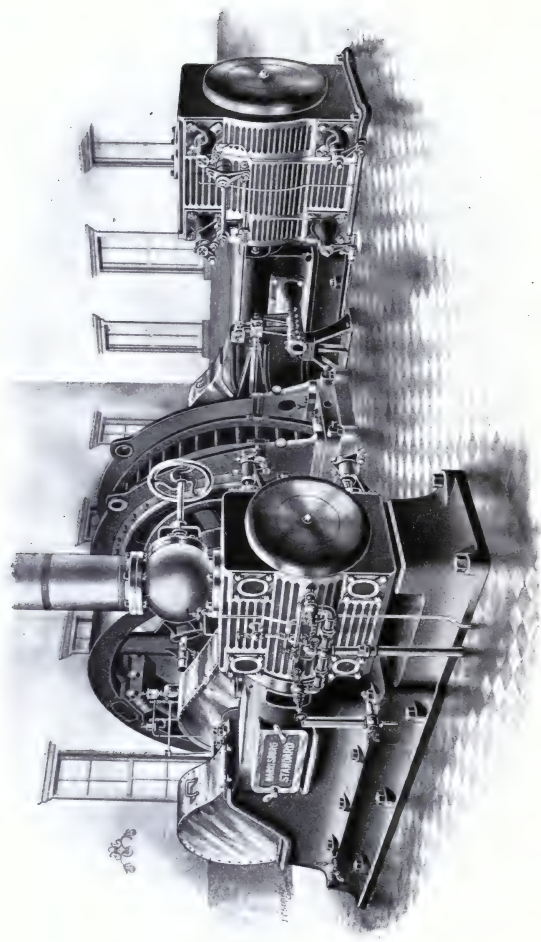


as would be the case if the cylinder and piston were perfect non-conductors of heat, the work done in expansion will be at the expense of the heat energy in the gas itself, and its temperature will therefore fall during the expansion. We have seen that the pressure is less as the temperature falls, other things being equal; hence under the conditions the pressure will fall faster than if the temperature were kept up by the addition of heat from outside. This is shown by the curve A of Fig. 9. Curves of this kind, representing expansion or compression without communication of heat to or from the gas, are called **adiabatics**. It is evident that adiabatic expansion along A from v_2 to v_1 is accompanied by a greater fall of pressure than isothermal expansion along I.



Both isothermal and adiabatic curves are of great importance





GENERAL CONSTRUCTION FLEMING CROSS COMPOUND FOUR-VALVE ENGINE DIRECT-CONNECTED STYLE, GENERAL VIEW.
Harrisburg Foundry and Machine Works.

in thermodynamic studies, but they represent conditions that are only imperfectly realized in practice. The general expression for an adiabatic curve is $PV^n = \text{constant}$. For air, $n = 1.405$. Most problems involving adiabatic and isothermal curves cannot be solved without the aid of higher mathematical processes than are used in this Paper.

Work Done in Expansion. Suppose we have a piston whose

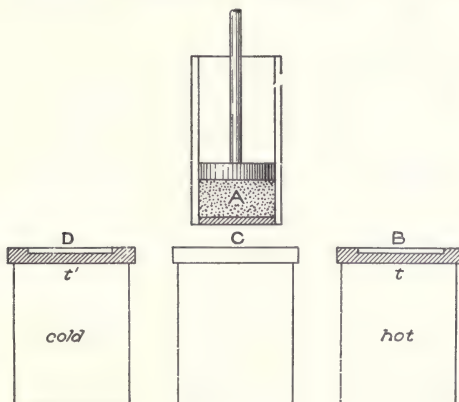


Fig. 11.

area is A square inches, which is acted upon by a pressure of p pounds per square inch, and which moves through a distance of m feet in consequence. Then the total pressure is pA pounds, and the work done is $pA \times m$ foot-pounds. But $A \times m$ is the volume of the cylinder swept out by the piston in its stroke; and calling this V , we have:

$$\text{Work done} = \text{pressure} \times \text{volume} = PV.$$

This can be conveniently shown on the pressure-volume diagram. Suppose B (Fig. 10) represents the pressure and volume of a gas, which then expands a little to the condition A. Then the average pressure during the expansion will be $\frac{1}{2}(Bb + Aa)$, and the work done will be $\frac{1}{2}(Bb + Aa) \times ab = \text{the area } BAab$.

Since we may regard the whole change from C to A as made up of portions like that from B to A, it follows that in changing from C to A along the path CBA the expanding gas will do the work represented by the area CAac.

The Carnot Cycle. This principle can be applied to the operation of heat engines. Let the working substance of a heat engine be a gas A, enclosed in a cylinder (Fig. 11) with non-conducting walls and piston and a perfectly conducting bottom. Let B be a hot body kept at the temperature t , and D a cold one kept at the temperature t' ; and let C be a nonconducting stand.

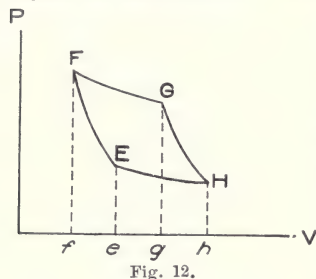
Then we may imagine the gas to undergo the following cycle of operations :

1. Set the cylinder on the stand C, with the gas at the temperature t' , and compress the gas adiabatically until its temperature rises to t . On the pressure-volume diagram (Fig. 12), this stage will be represented by the line EF, starting at E. The work done in compressing will be represented by the area EFfe.

2. Transfer the cylinder to the hot body B (Fig. 11), and allow the piston to rise by the expansion of the compressed gas. To maintain the temperature t during expansion, a certain quantity Q of heat must be supplied from B. This stage will be represented by the isothermal line FG (Fig. 12), and the work done by the expanding gas by the area FGgf.

3. Set the cylinder on the stand C (Fig. 11), and allow the gas to expand still further, until it cools to the temperature t' . This change is shown by the adiabatic line GH (Fig. 12), and the work done in the expansion by the area GHhg.

4. Place the cylinder on D and push the piston down to its first position. The heat Q' developed by the compression will be removed by the cold body D, and the temperature of the gas will remain unchanged. The change is shown by the isothermal line HE (Fig. 12), and the work done on the gas by the area HEeh.



The gas has now reached its initial condition. It has done an amount of work represented by the area FGH/f , and has had done on it the work represented by HEF/h . The difference $EFGH$ is the net work done in the cycle; and to do it, it has been necessary to take a quantity of heat Q from the hot body, and to discharge a quantity Q' into the cold body. The efficiency of the operation is

$$\frac{\text{Heat utilized}}{\text{Heat received}} = \frac{Q - Q'}{Q}.$$

The above cycle of operations is called the Carnot cycle, because Carnot first applied it as a method of reasoning. The efficiency of a steam engine working on the above cycle can be shown as follows:

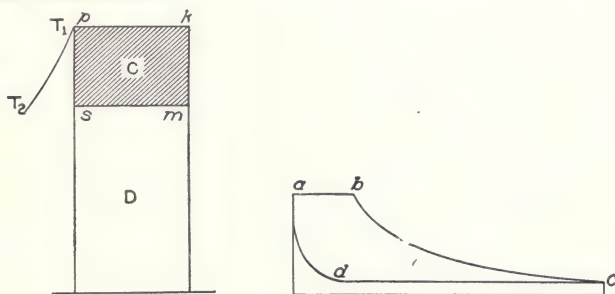


Fig. 13.

Instead of drawing a pressure-volume diagram whose vertical distances are pressures and whose areas represent work, let us draw one whose vertical distances are temperatures and whose areas represent the quantities of heat added during any change in the working fluid. A diagram of this kind is called a temperature-entropy diagram. Starting with a pound of water at T_1 (Fig. 13), let us convert it completely into steam at that temperature. This will be represented in our diagram by the line pk , and the heat added to produce the change will be shown by the areas $C + D$. Then let the steam expand adiabatically until it reaches the temperature T_2 . This is represented by the line km , which is vertical and represents no additional area, because no heat is added to or withdrawn

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from the steam from outside. Suppose, during the return stroke of the piston, the steam is cooled within the cylinder itself, until it arrives at the point *s*. In this operation we shall reject from the cylinder a quantity of heat represented by the area *D*. Finally, let us compress adiabatically the mixture of steam and water in the cylinder until we have once more a pound of water at the higher temperature T_1 . This gives the line *sp*, ending where the cycle began.

We have thus supplied during the cycle the heat represented by the areas $C + D$, and the heat rejected is represented by *D*. Therefore the efficiency is $\frac{C}{C + D}$, and from the diagram it will be seen that this is equal to $\frac{T_1 - T_2}{T_1}$.

In the actual engine the last step is very imperfectly performed, because only a small amount of steam remaining in the cylinder is compressed; the remainder, exhausted from the cylinder at the lower temperature T_2 , must be heated to the temperature T_1 by heat from the boiler, or replaced by an equal amount thus heated.

We thus have the important conclusion that the efficiency of a steam engine depends entirely on the ratio of the temperatures between which it operates; it follows that the efficiency is always small. For example, suppose an engine takes steam at 300°F and exhausts at 212°F . Its efficiency cannot be greater than

$$\frac{(300 + 461) - (212 + 461)}{(300 + 461)} = 11.6 \text{ per cent.}$$

EXAMPLE FOR PRACTICE.

Using the steam table on page 25, find the maximum possible efficiency of an engine taking steam at 150 lbs. absolute pressure and exhausting at 5 lbs. absolute pressure.

Ans. 23.9 per cent.

THE STEAM ENGINE.

Without entering into a detailed description of the mechanism, it will be sufficient here to say that in the steam engine, steam

is admitted under pressure from a boiler into a metal cylinder behind a piston, as represented in Fig. 14. Its pressure drives the piston forward, doing useful work. When the piston has moved through a part of its stroke the steam supply is cut off, and the stroke is completed by the expansion of the steam confined in the cylinder. By the first law of thermodynamics this expansion cools the steam, since work is done in the process; but the expansion is not adiabatic, since the cylinder and piston give up some heat to the steam within. At the end of the stroke the exhaust valve opens and the cooled steam escapes into the atmosphere or condenser through the exhaust pipe A. The operation is then repeated on the other side of the piston.

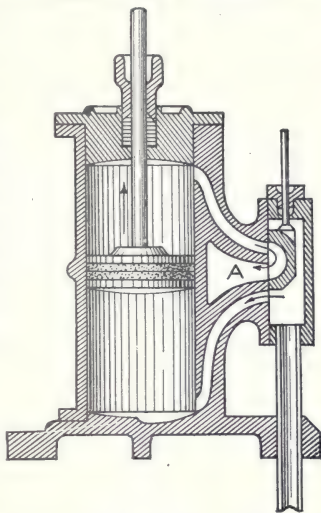


Fig. 14.

The pressure-volume diagram in Fig. 13 shows the process graphically, and can be instructively compared with the temperature-entropy diagram of the same figure. For purposes

of analysis it is immaterial whether we consider the steam to be supplied from outside, or whether we consider the whole operation of heating to take place inside the cylinder. The line pk , representing the conversion of water into steam at constant pressure, will appear on the pressure-volume diagram as the "admission-line" ab . The adiabatic line km is shown by the falling "expansion-line" bc , which shows the relation between pressure and volume after the steam is cut off. The line ms becomes the "exhaust-line" cd on our new diagram, representing the steam pressure while the steam is being pushed out of the cylinder. Near the end of the return stroke of the piston, the exhaust valve closes and the steam remaining in the cylinder is compressed by the piston. This process, nearly adiabatic, is

shown by the line da , corresponding to the line sp on the other diagram.

A pressure-volume diagram like the above is called an **indicator card**. It is very valuable in determining the power and performance of engines.

Compound Engines. Since the efficiency of a heat engine is evidently much improved by extending the range of temperature through which it works, and since we have also seen that it takes but little more fuel to produce a pound of high-pressure steam than a pound of low-pressure steam, it should now be clear that it is economical of fuel to use the highest pressures practicable, and to expand the steam as much as possible. But for certain reasons, discussed fully in the Instruction Papers on the Steam Engine, it is advisable to divide the expansion among two, three or even four cylinders, according to the initial steam pressure. Such engines are called compound, triple or quadruple-expansion, and are usually worked with a condenser. In such cases, while the first or high-pressure cylinder is intensely hot, the last or low-pressure cylinder is scarcely more than uncomfortably warm to the hand. The difference represents the heat spent in expansion, a part of which has gone to produce the useful work done by the engine, and a part of which is wasted.

The Hot-Air Engine. There are many forms of these machines, but their general principle varies little. A quantity of air is heated in an iron chamber over a fire, and then is allowed to expand behind a piston, doing work. At the end of the stroke the air is transferred, either by a pump or an auxiliary piston, to a cold chamber, kept cool by air or running water. On the next stroke the air, reduced in volume by its cooling, is forced by the pump into the hot chamber, where it is again heated and the cycle repeated.

Hot-air engines are economical in operation, but necessarily bulky for the amount of power produced. An examination of Fig. 13 will show why this must be so. The heat which must be supplied to the cylinder for every stroke of the piston is represented by the areas $C + D$, no matter what substance is used as a carrier. But since the heat capacity of air is very small compared with that of steam, the cylinder of the hot-air engine must

necessarily be much larger than that of the steam engine for equal power. For this reason hot-air engines are used only in comparatively small powers. The hot-air engine may indeed work between wider temperature limits, but this is not enough to offset the difference between steam and air as carriers of heat.

The Gas Engine. In the gas engine, so called, the energy is derived from the rapid combustion or explosion of a mixture of gas or gasoline vapor and air. In one form of engine the cycle of operations is as follows: A forward stroke of the piston draws into the cylinder a mixture of gas and air in such proportions as to make an explosive mixture. On the return stroke the "charge" is compressed. At or near the end of the stroke, the mixture is exploded, usually by a properly-timed electric spark, and the pressure within the cylinder rises to a high value. The piston is driven forward by the expansion of the hot gases, doing useful work at the expense of the heat-energy in them. At the end of the stroke the exhaust valve opens, and on the second return stroke the burnt gases are pushed out of the cylinder.

The above cycle is often called the **Otto cycle**. For engines working in this way there is thus only one working stroke in every four, and they must be provided with a very heavy fly wheel. Some engines are arranged to have every alternate stroke a working stroke. These are commonly called two-cycle engines, and are much used in propelling boats.

The gas engine has the thermodynamic advantage of working between very wide temperature limits, but is nevertheless subject to serious losses. It is not practicable to expand the exploded charge down to the atmospheric pressure; the gases are discharged while still possessing much available energy. This may be noticed in the sharp, barking exhaust from a gas engine unprovided with a muffler. But the most serious loss is in the transmission of heat to the cylinder walls. This loss is also present in the steam engine, but to a small extent may be recovered. In the gas engine, however, it is practically all wasted.

REFRIGERATING MACHINES.

The cooling produced by the evaporation of a volatile liquid has a very important application in refrigerating machinery.

Ammonia is generally used as the working substance, because it is cheap and satisfactory.

Fig. 15 shows the essential parts of a compression machine. The *compressor* A, kept cool by a jacket of running water, draws ammonia vapor through the valve *a*, compresses it highly and sends it through the valve *b* to the *condenser* B. This is a coil of pipe, also kept cool by running water; and it serves to condense the ammonia, which collects in the bottom coils. The valve D admits the liquid ammonia to the *vaporizer* C, which is also made

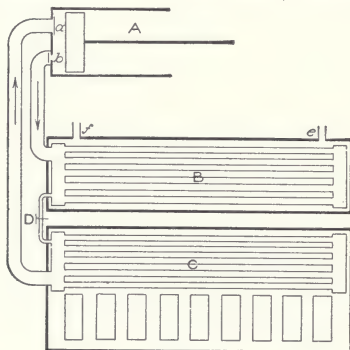


Fig. 15.

of pipe coils. In these it vaporizes and falls much below the freezing point of water. (It is well here to refer back to the experiment described on page 16.)

Many gases can be liquefied and used in this way; ammonia, carbonic acid and sulphurous anhydride (sometimes called sulphurous acid) in the liquid state, are regular articles of commerce.

In ice making, the vaporizer coils are immersed in a tank of strong brine. The water to be frozen is put into thin metal cans, which are then set into the cold brine, as represented in the figure.

In this way the heat liberated from the water in freezing is carried away by the ammonia vapor, and finally discharged into the cooling water circulating around the compressor and condenser. But since this is at a higher temperature than the source of the heat, the machine is not self-acting. It requires power to operate it, which is expended in compressing the vapor in A.

Instead of drawing the vaporized ammonia back into the pump cylinder, it may be absorbed by cold water, for which it has a strong affinity. Such a machine is called an *absorption machine*, and Fig. 16 shows the principle of a continuously operating absorption machine. The *generator* B contains a concentrated solution of ammonia in water, from which the ammonia is expelled by heat. The *condenser* C is a pipe coil, kept cool by running water, in which the ammonia condenses to the liquid state as soon as its pressure rises to the necessary value. The regulating valve V

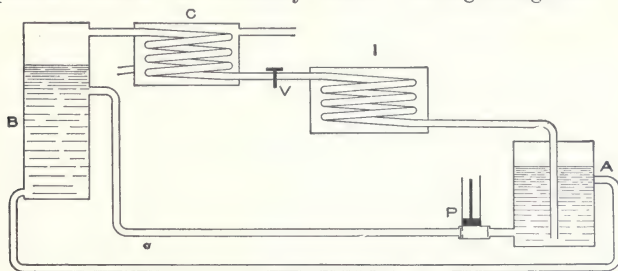


Fig. 16.

allows the condensed ammonia to escape to the *refrigerator* I, which corresponds to the vaporizer C of the compression apparatus. The *absorber* A is a tank of cold water in which the gaseous ammonia from I is absorbed. The pipes connecting A and B are arranged to take the most concentrated solution from A to B, and to return to A the water from which the ammonia has been driven. This is effected by the pump P.

In practice the generator B is placed over a furnace, and arrangements are also made for transferring heat from the hot liquid flowing from B to A into the cold liquid flowing from A to B.

LIQUEFACTION OF AIR.

That work done in compressing a gas heats the gas, is a fact familiar to every one who uses a bicycle pump. Similarly, the expansion of a gas against atmospheric or other pressure is accompanied by as decided a cooling. This may readily be observed by holding the finger in the jet of air escaping from a bicycle tire

valve. By suitable applications of this principle, it is possible to produce the most intense cold.

One of the simplest methods of liquefying air is shown in principle in Fig. 17. After thorough drying, the air to be liquefied enters through the pipe *a*, and in the compressor *C* is compressed to about 200 atmospheres (1 atmosphere = 14.7 pounds per square inch). *R* is a water cooler, to remove the heat of compression. The air thus cooled and strongly compressed passes

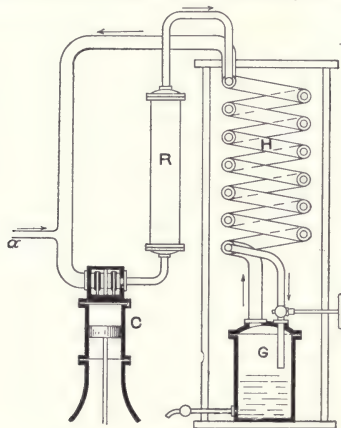
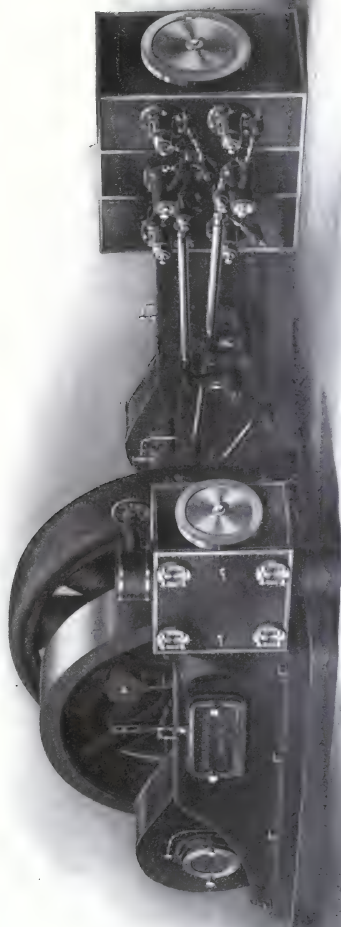


Fig. 17.

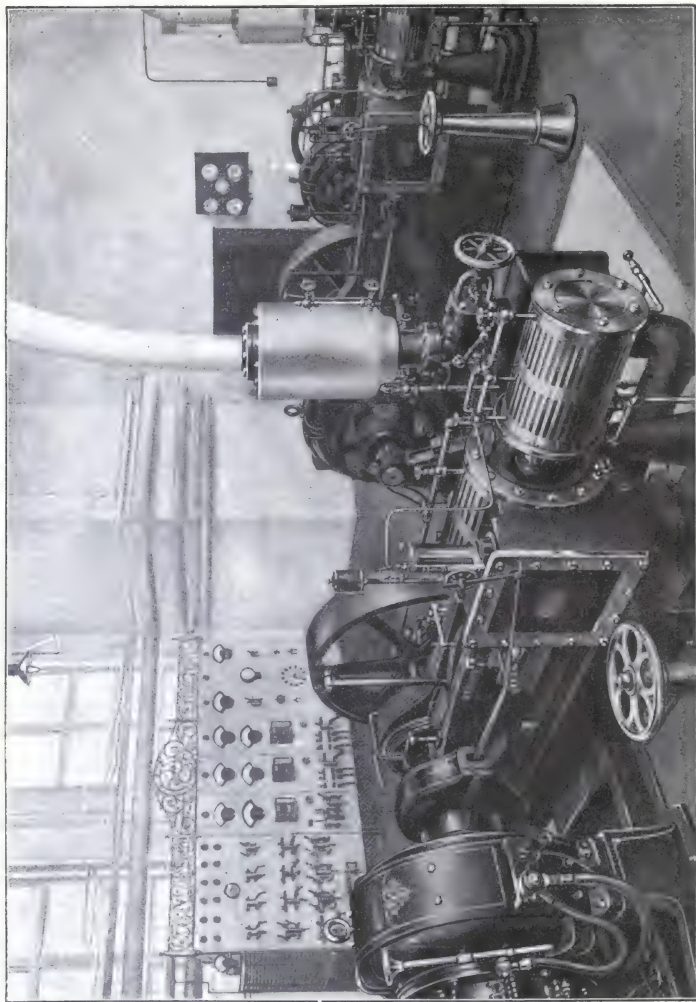
down through the inner tube of the helical coil *H* to a valve below.

Through this valve it escapes into the reservoir *G*, the expansion producing a considerable fall in temperature. The cold air then passes from the reservoir up through the outside tube of the helical coil, which surrounds the tube down which the air comes, thus cooling the compressed air in the inner tube. This cooled air is allowed to escape in its turn, becoming still colder by its expansion. As the process continues the temperature falls until liquid air begins to collect in the bottom of *G*, from which it may be drawn off.

With a 3-horse-power engine the yield is about a quart of liquid air per hour.



HORIZONTAL CROSS-COMPOUND SIDE-CRANK CORLISS-TYPE STEAM ENGINE
The Ball Engine Co., Erie, Penna.



INSTALLATION OF 125-H. P. TANDEM-COMPOUND AUTOMATIC STEAM ENGINES, DIRECT-CONNECTED TO GENERATORS
Skinner Engine Co., Erie, Penna.

THE STEAM ENGINE

PART I.

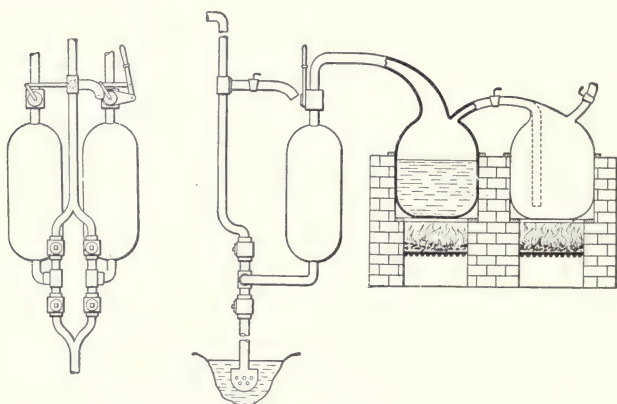
There are various kinds of engines from which mechanical work is obtained by the expenditure of heat. In the gas engine a mixture of gas and air is burned in the cylinder, the heat thus generated being converted into work by the expansion of the products of combustion. The action in oil and hot-air engines is very similar. The most important of all heat engines, however, is the steam engine, in which the heat in steam is transformed into work. It will be useful to review briefly some of the stages through which it has passed in its development.

The first steam engines of which we have any knowledge were described by Hero of Alexandria, in a book written two centuries before Christ. Some of them were very ingenious, but the best were little more than toys. From the time of Hero until the seventeenth century there was very little progress. At this time there began to be great need of steam pumps to remove water from the coal mines. In 1615, Salomon de Caus devised the following arrangement. A vessel, having a pipe leading from the bottom, was filled with water and then closed. Heat applied to the vessel caused steam to be formed, which forced the water through the pipe.

A little later an engine was constructed in the form of a steam turbine; but it was unsuccessful, and the attention of inventors was again turned to pumps.

Finally Thomas Savery completed, in 1693, the first *commercially successful* steam engine. It was very wasteful of steam as compared with our engines of today, but as being the first engine to accomplish its task it was a grand success. Savery's engine (Fig. 1) consisted of two oval vessels placed side by side and in communication with a boiler. The lower parts were connected by tubes fitted with suitable valves. Steam from the boiler was admitted to one of the vessels and the air driven out. The steam was then condensed and a vacuum formed by letting

water play over the surface of the vessel. When the valve opened, this vacuum drew water from below until the vessel was full. The valve was then closed and steam again admitted, so that on opening the second valve the water was forced out through the delivery pipe. The two vessels worked alternately. When one was filling with water, the other was open to the boiler and was being emptied. Of the two boilers, one supplied steam to the oval vessels and the other was used for feeding water to the first boiler. The second boiler was filled while cold and a fire lighted under it. It then acted like the vessel used by Salomon de Caus and forced a supply of feed water into the main boiler.



END VIEW.

Fig. 1.

SIDE VIEW.

A modification of Savery's engine, the pulsometer (Fig. 2), is still quite common. It is used in places where an ordinary pump could not be used and where extreme simplicity is of especial advantage. Its valves work automatically and it requires very little attention.

A serious difficulty with Savery's engine resulted from the fact that the height to which water could be raised was limited by the pressure which the vessels could bear. Where the mine was very deep it was necessary to use several engines, each one raising the water a part of the whole distance. The consumption of coal

in proportion to the work done was about twenty times as great as that of a good modern steam engine. This was largely, though not entirely, due to the immense amount of steam which was wasted by condensation when it came in contact with the water in the oval vessels.

The next great step in the development of the steam engine was taken by Newcomen, who in 1705 succeeded in preventing contact between the steam and the water to be pumped, thus diminishing the amount of steam uselessly condensed. He introduced the first successful engine which used a piston working in a cylinder.

In Newcomen's engine, shown in Fig. 3, there was a horizontal lever pivoted at the center and carrying at one end a long heavy

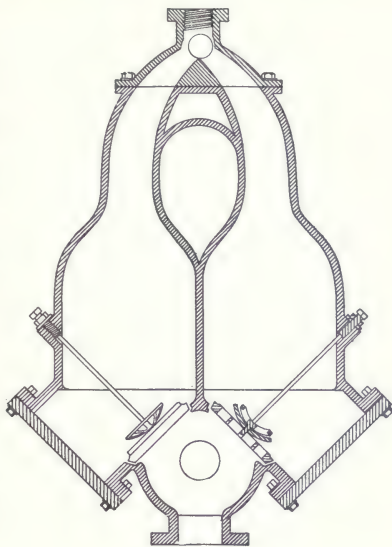


Fig. 2.

rod which connected with a pump in the mine below. A piston was hung from the other end of the lever, and worked up and down in a vertical cylinder, which was open at the top. Steam acted only on the lower side of the piston. Steam at atmospheric pressure was admitted from the boiler to the cylinder, and as the pressure was the same both above and below the piston, the falling of the heavy pump rod raised the piston. A jet of water was now passed into the cylinder to condense the steam and form a vacuum. This left the piston with atmospheric pressure above and very slight pressure below, so it was forced down and the pump rod again raised. Steam could again be admitted to the cylinder, the pump rod would fall, and so on indefinitely.

In the days of Newcomen it was very difficult to obtain good workmanship. For this reason it was often necessary to make the cylinders of wood, and even then there might be a space of one-eighth of an inch between the wall of the cylinder and the piston. In order to prevent steam from blowing through this passage, or air from leaking in when the steam was condensed, it was customary to keep a jet of water playing on the top of the piston.

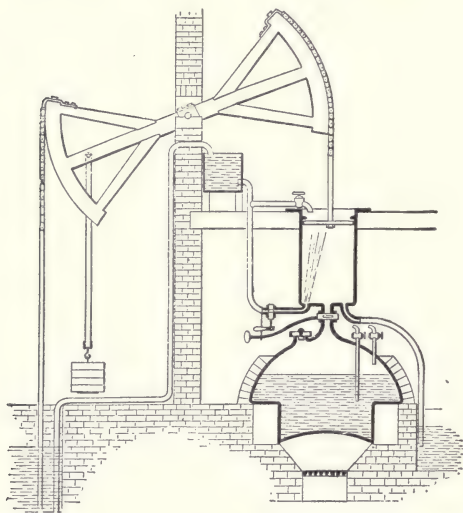
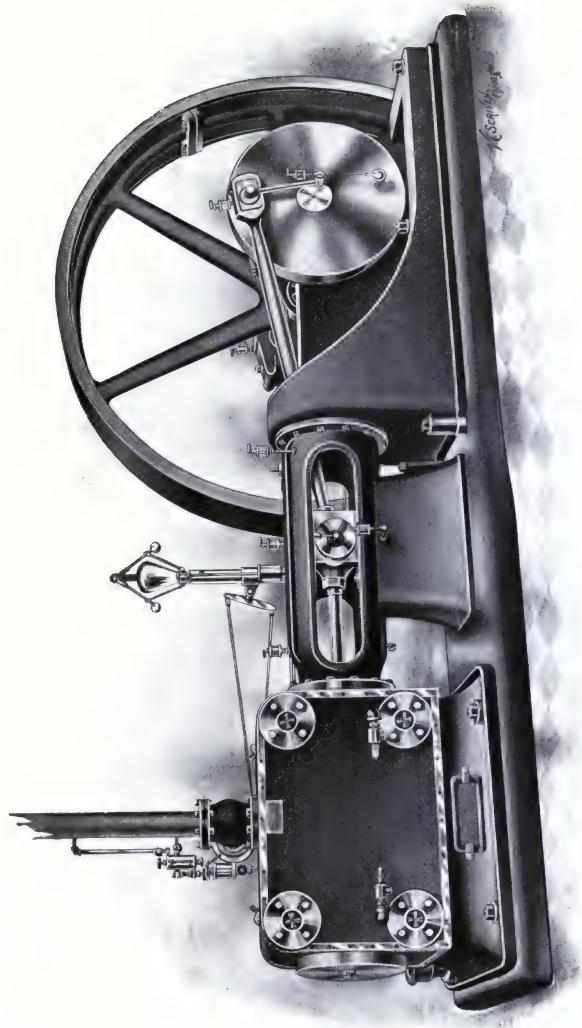


Fig. 3.

One great trouble with all these engines was that they required some one to open and close the cocks. Boys were generally employed to do this work. In order to get time to play, one of them rigged a catch at the end of a cord which was attached to the beam overhead. This did the work for him. Making the valves automatic in this way made it possible to dispense with the services of the boy and at the same time greatly increase the speed of the engine. This engine was improved slightly from time to time by different inventors and was very extensively used until



MURRAY TANGYE FRAME CORLISS ENGINE FOR BELT DRIVE

Watt's time. Some of them are in existence today. While this engine was a success and a great improvement over its predecessors, it was still very large, wasteful and heavy, in comparison with the work done. When the cylinders were made of iron they were simply cast and not bored, thus leaving a rough, stony coating over the iron, called the skin.

In the year 1763, a small model of a Newcomen engine was taken to the shop of an instrument maker in Glasgow, Scotland, to be repaired. This instrument maker, whose name was James Watt, had been studying steam engines for some time and he became very much interested in this model. He was a man of great genius, and before he died his inventions had made the steam engine so perfect a machine that there has been but one really great improvement in it since his time; namely, compound expansion. All other improvements have been merely following in the line of his suggestions and constructing what he could not for lack of good tools.

He found that to obtain the best results it was necessary, "*First, that the temperature of the cylinder should always be the same as that of the steam which entered it; and, secondly, that when the steam was condensed it should be cooled to as low a temperature as possible.*" All improvements in steam-engine efficiency have been in the direction of a more complete realization of these two conditions.

In order to keep the cylinder nearly as hot as the entering steam, Watt no longer injected water into the cylinder to condense the steam, but used a separate vessel or condenser. He made his piston tight by using greater care in construction, so that it was not necessary to have a water seal at the top. He then covered the top of the cylinder to prevent air from cooling the piston. When this was done he could use steam above as well as below the piston; this made the engine double acting.

Also, in the effort to keep the cylinder as hot as the entering steam, he enclosed the cylinder in a larger one and filled the space between with steam. This was not often done, however, and only of late years has the steam jacket been of much advantage. He also used steam expansively, that is, the admission of steam was stopped when the piston had made a part of the stroke; the rest

of the stroke was completed by the expansion of the steam already admitted. This plan is now used in all engines that are built for economy.

Other inventions made by Watt on his steam engine were : a parallel motion, that is, an arrangement of links connecting the

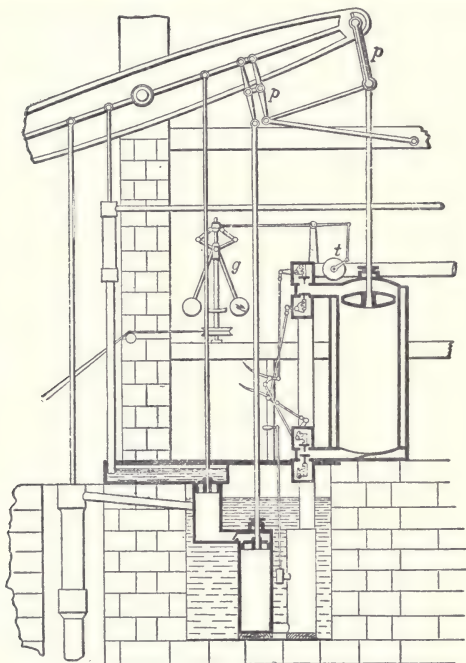


Fig. 4.

end of the piston rod with the beam of the engine in such a way as to guide the rod almost exactly in a straight line; the throttle valve, for regulating the rate of admission of steam and the centrifugal governor, which controlled the speed by acting on the throttle valve.

Watt also invented the "indicator," by means of which diagrams are made which show at all points the relation between the

pressure in the cylinder and the position of the piston at that instant. His assistant, Murdoch, invented the slide valve as a means of admitting and releasing the steam. Fig. 4 shows Watt's final engine.

Watt, like other early inventors, sold many of his engines to miners, who had been using horses to pump out the mines, and for this reason he rated his engines by the horse-power. Although this term has an historical derivation it has no real significance, and no relation whatever to the power of a horse. It is an established unit for measuring the rate at which work is done. One horse-power is the amount of work necessary to raise 33,000 pounds through one foot in one minute; and we may say that one horse-power is equal to 33,000 foot pounds per minute.

Watt saw that by using high-pressure steam he could get more work from it; but as it was not possible to make very reliable boilers he never used a pressure of more than seven pounds per square inch above the atmosphere. About the year 1800 comparatively high pressures came more into use and the *non-condensing* engine was introduced. In Watt's engine, and all those preceding his, a vacuum was produced in front of the piston by condensing the steam, and either the atmosphere or steam at atmospheric pressure pushed it through the stroke. In the non-condensing engine, using high-pressure steam, the space in front of the piston could be opened to the atmosphere at exhaust, and although the atmospheric pressure resisted its motion the pressure of the steam behind the piston was still greater than that of the air. These engines were much more simple than the condensing engines, as they required no condenser.

About this time what would now be called a compound engine was introduced by Hornblower and later by Woolf. It had two cylinders of different size. Steam was admitted into the smaller cylinder, and then passed over into the larger. The steam expanded a little in the smaller cylinder and much more in the larger one.

A great many attempts were made to build *locomotives*, but they were generally unsuccessful until George Stephenson built his engine, the "Rocket," in 1829. The principal new feature of this engine was the improved *steam blast* for increasing the

draft in the furnace and so making possible the use of a smaller boiler. Later he used the "link motion," which enabled the engine to be quickly reversed and the amount of expansion varied.

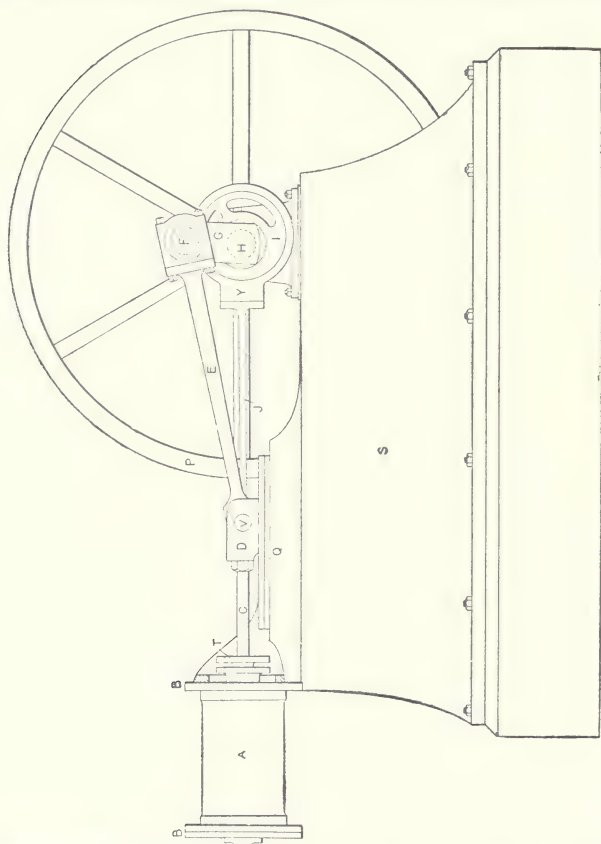


Fig. 5.

The Stephenson link motion may be seen on almost any locomotive. It is simply a device by which either of two eccentrics may be made to move the valve.

About the year 1814, Woolf introduced a compound pump-

ing engine in the mines of Cornwall, but a simpler engine was later introduced and Woolf's engine fell into disuse. This later engine became known as the Cornish Pumping Engine and was famous for many years because of its economy. It was the first engine ever built that could compare at all with modern engines in the matter of steam consumption. It consisted of a single cylinder placed under one end of a beam from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted on the upper side of the piston for a short portion of the stroke and allowed to expand for the remainder of the stroke. This forced the piston down, lifted the heavy pump rod and filled the pumps with water. Then communication was established between the upper and under side of the piston, exhaust occurred, and the heavy pump rod fell, lifting the piston and forcing the water out of the pumps. The cut-off was about .3 stroke, and the pump made about seven or eight complete strokes per minute with a short pause at the end of each stroke to allow the valves to close easily and the pumps to fill with water. The cylinder was jacketed. These engines needed great care and were in charge of competent men, to whom prizes were frequently given for the best efficiency, which doubtless accounts for their wonderful performance.

PARTS OF THE STEAM ENGINE.

Fig. 5 shows the elevation of a simple form of steam engine.

The Cylinder A (see plan, Fig. 6) is that part of the engine in which the piston moves back and forth. It is made of cast iron and accurately bored. Great care must be taken in this work, for any unevenness will allow steam to leak through between the piston and cylinder walls or it may even cause the piston to stick or work hard. In large engines the cylinder consists of two parts, the outer or cylinder proper and a comparatively thin cast iron liner. A space can be left between them for a steam jacket. Should the cylinder liner be damaged, it can be replaced without the expense of a new cylinder.

The Cylinder Heads B cover the ends of the cylinder and are securely bolted thereto. In the crank-end cover there is an opening for the piston rod to pass through. This opening is made

steam tight by a stuffing box which surrounds the piston rod. Sometimes the piston rod is prolonged beyond the piston and through the front cover. This extension of the piston rod is to

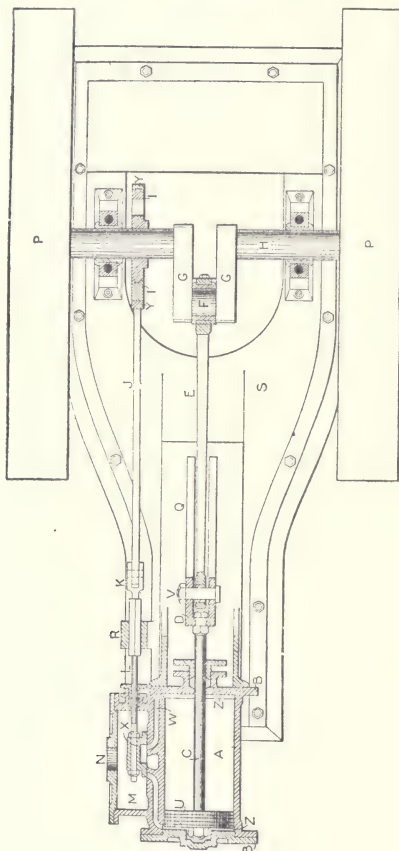


Fig. 6.

help steady the piston in a long stroke and is known as the tail rod. When a tail rod is used another stuffing box must also be provided for the head-end cover.

The Piston U, in small engines, is usually a thick disc of iron or steel, as shown in Fig. 6. It is often made conical, as shown in Fig. 7, to better withstand the steam pressure and to gain space. The piston must fit the cylinder steam tight and yet move easily. To accomplish this, one or more grooves in the piston are filled with packing (usually metallic), or spring rings may be used.

The Piston Rod C (Figs. 5 and 6) is made of steel and connects the crosshead and the piston to which it is rigidly fixed.

The Crosshead D serves to join the piston rod and connecting rod. At one end it is fastened to the piston rod, and at the other end is the wrist pin V on which the connecting rod swings. It is guided to and fro by the crosshead guides Q.

The Connecting Rod E is a steel forging from three to eight times the length of the crank, depending upon the type of engine. One end is jointed to the crosshead by the pin V, called the wrist pin, while the other encircles the crank pin and revolves with it. A detail view of one end is shown in Fig. 8; the other end is frequently similar. In some cases the small end is forked, as shown in Fig. 9.

The Crank Pin F forms the connection between the crank and connecting rod.

The Crank G, equal in length to one-half the stroke of the piston, converts the back and forth motion of the connecting rod into circular motion. It may be simply an arm, as shown in Fig. 10, or a complete disc keyed to one end of the shaft, as shown in Fig. 11. The disc is more nearly balanced than the crank.

The Shaft H transmits the rotary motion from the crank to the fly wheel P.

The Frame of the engine S is a heavy casting, which supports the cylinder and bearings. It should be securely bolted to the foundation.

The Steam Chest M receives steam directly from the boiler, and the steam passes thence through the ports W into the cylinder.

The Eccentric I is a disc keyed to the shaft so that its center and the center of the shaft do not coincide. The eccentric strap Y encircles the eccentric and imparts a reciprocating motion to the valve stem L and the eccentric rod J. This action is similar to

that of the crank and connecting rod, but exactly reversed. K is the valve stem crosshead and R its guides.

The Slide Valve X is the valve for alternately admitting

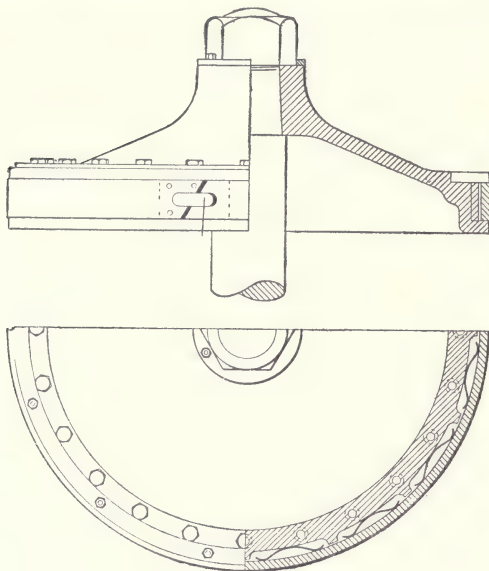


Fig. 7.

steam to the cylinder and releasing it. It has a cup-shaped cavity in its face through which the exhaust steam passes. It is situated in the steam chest and is moved by the valve gear, that is, the eccentric and the eccentric rod.

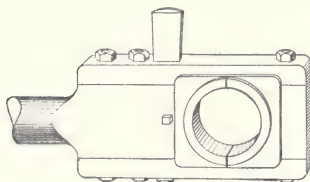


Fig. 8.

The Clearance Z is the space between the piston and the cylinder head (when the piston is at the end of the stroke), together with the volume of the steam ports. This volume must

be filled with steam before the piston can start. It is usual to

express the clearance as a certain per cent of the volume swept through by the piston.

The crank may revolve in either direction. If we stand by the cylinder, facing the crank shaft, and the crank moves away from us as it passes over the shaft, we say that it is running *over*. If it moves away from us as it passes under the shaft, we say that the engine is running *under*. The action of steam in the

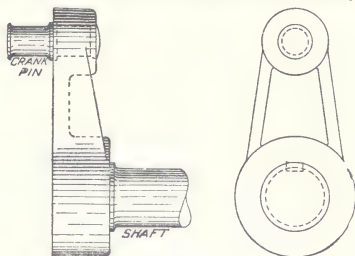


Fig. 10.

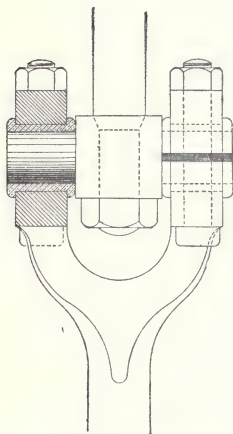


Fig. 9.

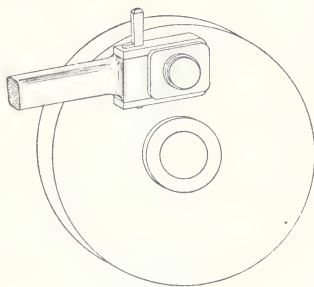


Fig. 11.

cylinder of an engine is very complicated, and its discussion will be taken up later in the course.

TYPES OF ENGINES.

Classification. There are so many different types of engines that it is difficult to classify them all properly. Most engines belong to several classes at one and the same time. For instance, there are condensing and non-condensing engines; there are

simple, compound, triple and quadruple expansion ; there are high speed and low speed, vertical and horizontal, locomotive, stationary

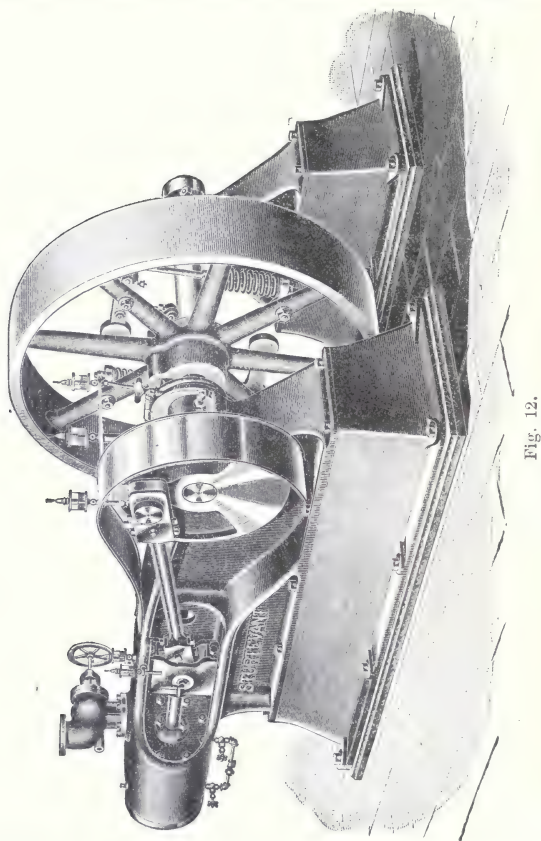


Fig. 12.

and marine, and many other classes into which these might be further subdivided.

Simple Engines. The simplest type of engine is the simple expansion. It has one cylinder and admits steam for a part of the stroke, expands it during the remainder and exhausts either into

the atmosphere or into a condenser. Simple engines (see Figs. 5 and 12) are now used only for comparatively small powers, say 100 H. P. or less, and although more extravagant of fuel than the others, may still be the most economical financially if low first cost is an important item, if they are not run continuously, or if the load fluctuates widely.

Compound Engines have two cylinders known as the high pressure and low pressure. Steam enters the smaller or high pressure cylinder and then expands until release, when it is exhausted into the larger cylinder, where the expansion is finished. The cylinders should be so proportioned that approximately the same amount of work can be done in each. The first cylinder is small, because it has the higher steam pressure, and a given weight of steam occupies less space when at high pressure. The second must be large, so that the volume at cut-off can contain all of the steam exhausted from the high.

Besides being more economical the compound has a distinct mechanical advantage. The two cranks may be set at right angles, so that when one is on dead center the other is at a position of nearly its greatest effort. This makes a dead center impossible, and gives a more uniform turning moment. Then the individual parts may be made lighter, and are thus more easily handled, but the engine is much more costly, and it is nearly twice as much work to take care of it.

When the cranks of a compound engine are at 90° , the low-pressure piston is not ready to receive steam when the high pressure exhausts, hence there must be a receiver to hold the steam until admission occurs in the low. Such engines are called *cross compound*. Fig. 13 shows one form. Sometimes instead of having the cranks at 90° they are placed together or opposite. Then the strokes begin and end together, and the high can exhaust directly into the low without a receiver. Such engines are called *Woolf engines*. A tandem compound engine, shown in Fig. 14, has both pistons on one rod, the high-pressure piston rod forming the low-pressure tail rod. Such engines are less expensive because there is but one set of reciprocating parts instead of two, but like simple engines they have the disadvantage of dead points.

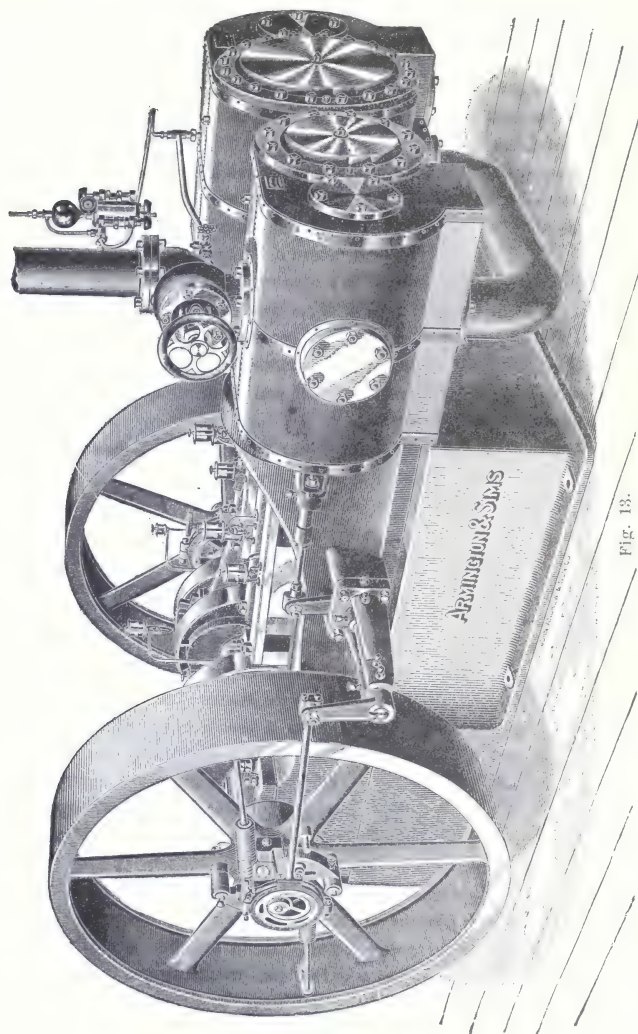


Fig. 13.

Triple Expansion Engines expand the steam in three stages instead of two. There are usually three cylinders, the high, intermediate, and low, arranged with cranks 120° apart. This gives a more uniform turning moment than a compound. Sometimes there are four cylinders to the triple, one high, one intermediate, and two low. This arrangement gives better balance and is often used in marine work.

For triple engines there must be a receiver between each two cylinders. Fig. 15 shows the essential features of a triple expansion engine.

Quadruple Engines expand their steam in four stages instead of three. Multiple-expansion engines are nearly always condensing.

Cylinder Ratios. There are several considerations to be remembered when proportioning the cylinders of multiple-expansion engines. The ratio of the cylinders should be such that each develops nearly the same power; the drop in pressure between the cylinders and receivers should be small, and the strains in the cylinders about equal.

There are many formulas in use, some simple, others involving mathematical calculation. A common rule for compound engines is to make the ratio of the cylinders equal to the square root of the total ratio of expansion. Thus if the steam has a ratio of expansion of 9, the ratio of the cylinder volumes will be $\sqrt{9} = 3$, or the low-pressure cylinder will have a volume 3 times as great as the high-pressure cylinder. If the cylinder ratio is 3, and the length of stroke is the same for both, the diameter of the low-pressure cylinder will be 1.75 times that of the high-pressure cylinder.

Another rule is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fractional part of the stroke completed when cut-off occurs in the high-pressure cylinder.

Suppose the ratio of expansion is 9, as above, and that cut-off occurs at $\frac{1}{3}$ of the stroke in the high-pressure cylinder. The ratio of cylinder volumes will be $9 \times \frac{1}{3} = 3$. If cut-off occurs at $\frac{1}{2}$ the stroke, the ratio will be $9 \times \frac{1}{2} = 4.5$.

For triple expansion engines the low pressure cylinder is made large enough to develop the whole power if steam at boiler pressure is used.

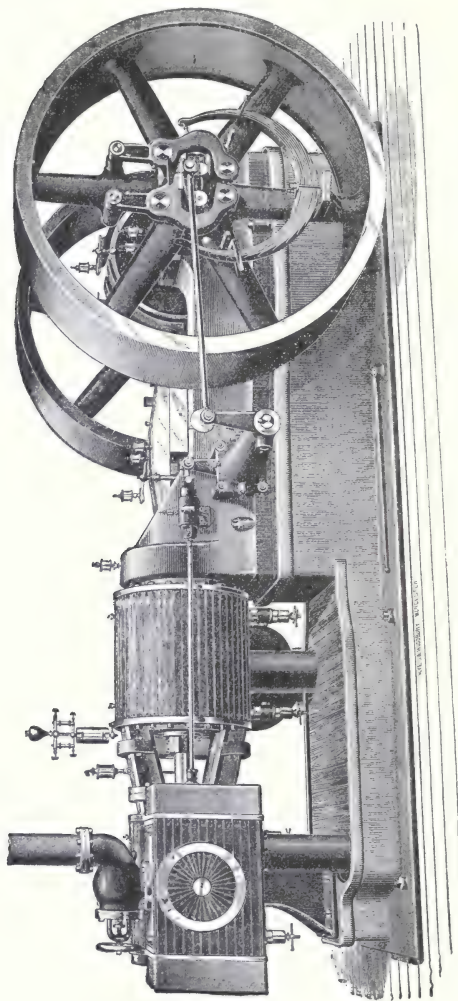


Fig. 14.

The intermediate cylinder is made approximately of a mean size between the high and the low. The area of the intermediate piston is found by dividing the area of the low by 1.1 times the square root of the ratio of the low to the high.

We may write the above thus,

$$\begin{aligned}\text{Area of high-pres. cyl.} &= \frac{\text{Area of low-pres. cyl.}}{\text{Cut-off of high-pres.} \times \text{ratio of exp.}} \\ \text{Area of inter. cyl.} &= \frac{\text{Area of low-pressure cylinder.}}{1.1 \times \sqrt{\text{ratio of low to high.}}}\end{aligned}$$

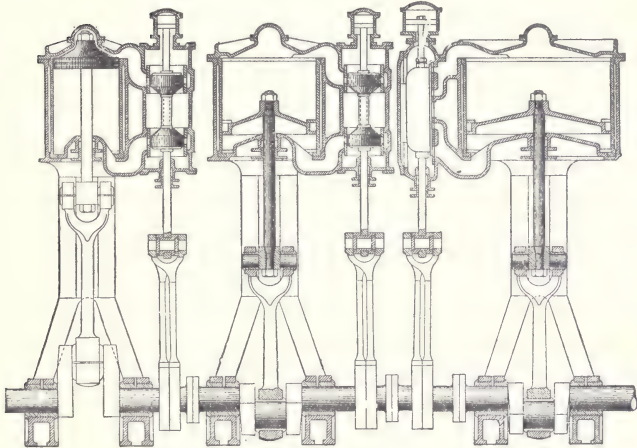


Fig 15.

In general the volumes for triple expansion are of about the following ratios, 1 representing the volume of the high pressure cylinder:

$$1 : 2.25 \text{ to } 2.75 : 5 \text{ to } 8.$$

For quadruple-expansion engines the ratios are as follows:

$$1 : 2 \text{ to } 2.33 : 4 \text{ to } 5 : 7 \text{ to } 12.$$

High Speed Engines. Of late years there has been a demand for engines of higher speeds than were formerly used. It was found desirable to run dynamo-electric machines by connecting them *directly* to the shaft of the engine rather than by belts as before.

This required engines running from 200 to 1,000 revolutions per minute instead of from 60 to 90 revolutions. Also for engines in

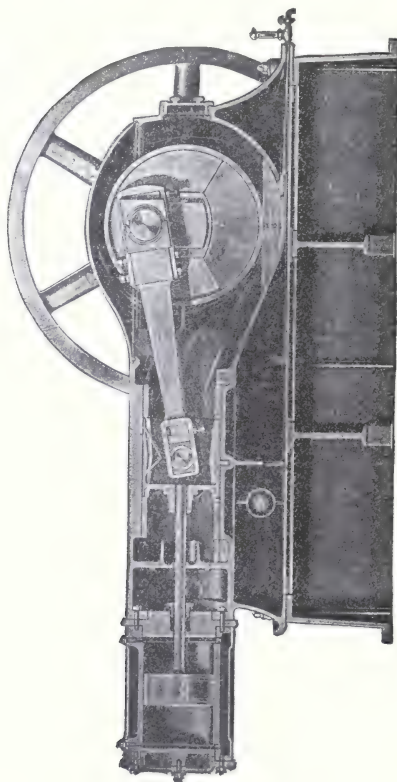


Fig. 16.

torpedo-boats, speeds as high as 400 or 500 revolutions are common.

Running at high speed requires various changes. Reciprocating parts must be made lighter to reduce the vibration, and must be



HORIZONTAL CROSS-COMPOUND SINGLE-VALVE ENGINE, DIRECT-CONNECTED TO GENERATOR

more carefully proportioned to maintain balance. Bearings must be made very much larger to reduce the pressure in order that the friction may not be excessive and cause heating. Special care is necessary that bearings should be tight, since the least looseness will cause knocking and hammering until the bearing is ruined. Parts must be as simple as possible and so arranged as to need the least possible attention. In slow-speed engines the engineer can watch the oil cups and oil any part while it is running. But this is impossible in high-speed engines, and special means must be used to insure plenty of oil to the bearings and the cylinder.

These peculiarities of high-speed engines may be easily seen in any engine for electric lighting, for running fans, etc. It is also necessary that the speed of engines for running electric generators should be very steady, as the slightest change in the speed make the lights flicker. The fly wheels are, therefore, larger or heavier than for other types and the governors are made specially sensitive. Fig. 16 shows a horizontal high-speed engine with the working parts encased. Fig. 17 shows a vertical high speed.

For any *double-acting* engine, that is, for any engine in which the steam acts first on one side and then on the other side of the piston, the piston first pushes and then pulls on the connecting rod and crank. At each half revolution of the crank the direction of the pressure reverses. It is this change of pressure which causes the pounding if the bearings are at all loose. This is one of the greatest troubles with high-speed engines. In order to avoid these rapid reversals in pressure, *single-acting* vertical engines are used to a considerable extent. In such engines the steam is admitted only to the *head end* of the cylinder. The other end stands open. The connecting rod is in compression throughout the whole revolution instead of being first in compression and then in tension. Besides insuring that the piston shall always push, this arrangement simplifies the valves.

There are several good single-acting, high-speed engines. One of the earliest made was introduced by Brotherhood. He used three cylinders, set around the shaft 120° apart. Another well-known example is the Willans "central valve" engine. These are both English engines. A well-known American engine

of this type is the Westinghouse high-speed engine, a section of which is shown in Fig. 18.

Vertical and Horizontal Engines. At the present time the most common type of engine is the horizontal direct-acting, that

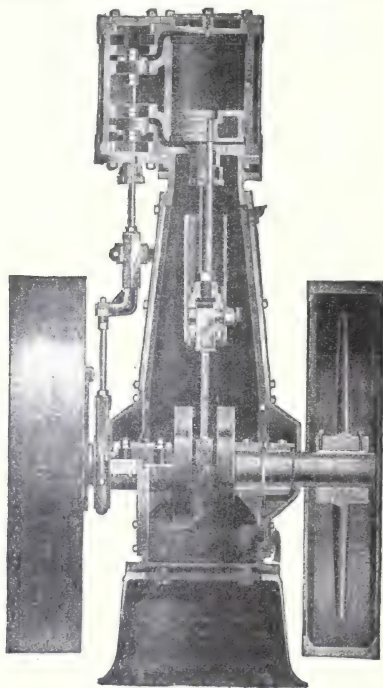


Fig. 17.

is, an engine whose cylinder is horizontal and whose piston acts on the crank through a piston rod and connecting rod. In small engines the whole is often on one bed plate. Such engines are called self contained. The cylinder is either bolted to the back of the bed plate or rests directly on it.

In marine work vertical engines are used in almost every case. The reason for this is, of course, the saving of floor space, which is so important in a vessel. This saving of space, however, is also

very important in many cases such as in crowded engine rooms of cities where land is expensive, and as there are a number of advantages which vertical engines have over horizontal, they are coming largely into use in stationary practice.

A second advantage of the vertical over the horizontal engine is the reduction of cylinder friction and unequal wear in the cylinder of the latter. In the horizontal engine the piston is generally supported by resting on the cylinder, which is gradually worn until it is no longer round. This causes leakage of steam from one side to the other. Evidently this is entirely avoided in the vertical engine.

Still another advantage of the vertical engine is the greater ease of balancing the moving parts so that there shall be no jarring or shaking. It is impossible to perfectly balance a steam engine of one or two cylinders. If it is balanced so that there is no tendency to shake sidewise, it will shake endwise; and if it is balanced endwise it will shake sidewise. The jarring is due to the back and forth motion of the reciprocating parts and the centrifugal force of the crank and connecting rod. The crank can be readily balanced by making it extend as far on one side of the shaft as it does on the other, but the piston and connecting rod are more difficult to balance. We can greatly reduce the effect of jarring if we balance the crank and make the endwise throw come in line with the foundation, which should be heavy enough to absorb the vibration transmitted. In a horizontal engine this endwise throw not being in line with the foundation will cause vibration in the engine itself.

In machines that can be anchored down to a massive foundation a state of defective balance only results in straining the parts and causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is of much more consequence in locomotive and marine engines.

To sum up the general advantages of vertical engines: they have less cylinder wear, they take up less floor space, and they can be better balanced. In addition to these there are certain advantages which vertical engines have for certain kinds of work.

The disadvantages of vertical engines are as follows: The

pressure on the crank-pin is greater during the *down* stroke than during the *up* stroke because, during the down stroke the weight of the reciprocating parts is added to the steam pressure, and during the up stroke this weight is subtracted.

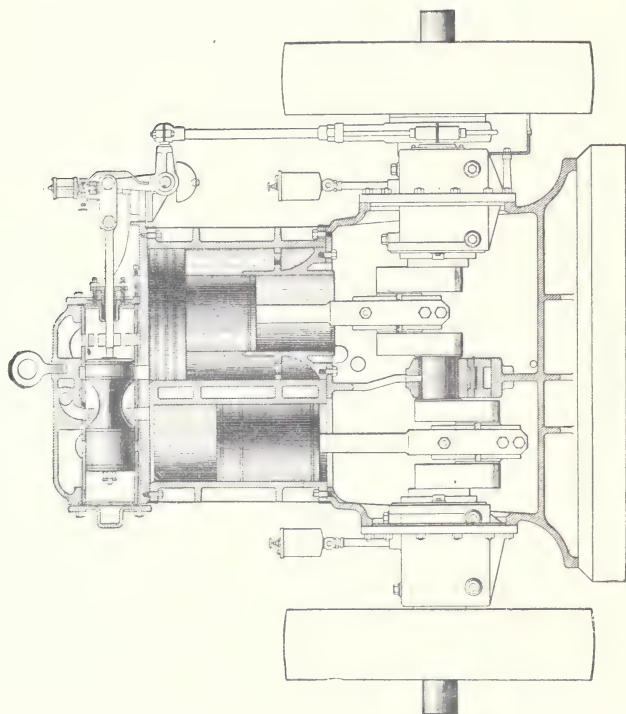


Fig. 18.

Another difficulty is that in large engines the various parts are on such different levels that they require considerable climbing. This requires more attendants and is sometimes the cause for neglect of the engine.

The foundations for vertical engines generally need to be *deeper* than those for horizontal engines. At the same time, however, they need not be as *broad*.

Marine Engines. The first steam vessels were fitted with paddle-wheels, and as beam engines were the most common, this form of engine was used. Its construction, however, was somewhat modified for this service. This arrangement of beam engine and paddle-wheel was used for many years and was applied to ocean vessels as well as to small river boats. It is still used, especially in this country, on river steamers and some coast steamers. The beam is supported by large A frames on the deck, and the engines are about on a level with the shaft.

Engines of this type take up rather more room than those now in common use, partly because of great size, and also because of the shaft and paddle-wheels. Another disadvantage is that in heavy weather, when one paddle-wheel is thrown out of water the other is deeply immersed and takes all the strain, so that there is a tendency to rack the boat. Then again if the boat is loaded heavily the paddle-blades are very deeply immersed while if light they barely touch the water. It is hard to handle the engines satisfactorily under both conditions.

The introduction of the screw propeller overcame these difficulties very largely and at the same time required a much faster running engine. At first, the increased speed was supplied by the use of spur-wheel gearing, but gradually higher speed engines were built and connected directly to the propeller shaft. It was, of course, difficult with small width at each side of the shaft to use horizontal engines, therefore various arrangements of inclined engines were used before the vertical engine was finally chosen by all as the standard form for marine work. It is only in recent years that the vertical engine has become general in naval work and in merchant steamers.

In merchant ocean steamers the common form has three cylinders set in line, fore and aft, above the shaft. The cranks are 120° apart, to give a very even turning moment. The three cylinders are worked *triple expansion*. The valves are usually piston valves on the high and intermediate, and double ported slide valves on the low. Sometimes piston valves are used on all

the cylinders. Plain slide valves are not suitable for high-pressure work of any kind.

For engines on ocean vessels it is necessary to use surface condensers in order that the same water may be used over and over again. If it were necessary to take in sea-water for the boilers they would soon become clogged with the salt and require cleaning. Surface condensers for marine work are generally made up of a large number of brass tubes of from $\frac{3}{4}$ inch to 1 inch in diameter. In some cases the cold water is forced to flow through the tubes while the steam comes in contact with the outside of the tubes.

In any marine plant there are four special pumps. The *first* is the air pump for the condenser. This is usually made large so that in case there is a leak in the condenser it can take charge of the water even if it becomes necessary to run as a jet instead of a surface condenser. The *second* is the feed pump for the boilers. The *third* is the circulating pump, which forces the current of cold water through the condenser. The last is the bilge pump, which pumps out water that gathers in the bilge of the ship by leakage or otherwise. In case of a serious leak all the pumps can be made to pump from the bilges. In some old types all these pumps were worked from the main engine; generally, however, the feed pump and the circulating pump are separate, as also the bilge pump. The circulating pump is, in many modern engine rooms, of the centrifugal type.

Locomotive Engines. Of all steam engines the most inefficient is the steam locomotive. In the first place, the boiler must be forced so hard that the products of combustion pass off at a very high temperature and consequently carry away a great deal of heat. Bits of entirely unburned or partially burned coal are drawn through and wasted.

In the second place, the boiler is exposed to great loss of heat by radiation. Although its surface is lagged, it cannot be very effectively covered, and the wind takes away a great deal of heat.

Mechanically also the locomotive is very imperfect. In most good steam engines the efficiency, that is to say, the ratio of the *effective horse-power* to the *developed horse-power* is fully $\frac{9}{10}$ or 90 per cent. In the locomotive this ratio was shown to be about 43

per cent by two independent tests. This is in part due to the special difficulties in locomotive construction, but the principal losses are those caused by radiation and the escape of heat from the stack.

As to locomotive boilers, Mr. Forney says, "The weight and dimensions of locomotive boilers are in nearly all cases determined by the limits of weight and space to which they are necessarily confined." It may be stated generally that within these limits a locomotive boiler cannot be made too large. In other words, boilers for locomotives should always be made as large as possible under the conditions that determine the weight and dimensions of the locomotives.

There are certain types of locomotives common in American practice which have special names. The eight-wheel or "American" passenger type of locomotive has four coupled driving-wheels and a four-wheeled truck in front. The "ten-wheel" type has six coupled drivers and a leading four-wheel truck. This type is used for both freight and passenger service. The "Mogul" type is used altogether for freight service; it has six coupled drivers and a two-wheel or pony truck in front. The "Consolidation" type is used for heavy freight service. It has eight coupled driving-wheels and a pony truck in front. There are also a great many special types for special purposes. In switch-yards a type of engine is used which has four or six drivers with no truck. The Forney type has four coupled driving-wheels under the engine and a four-wheel truck carrying the water-tank and fuel. This type is used on elevated roads largely. "Decapod" engines are a type used for heavy freight service, having ten coupled driving-wheels and a two-wheel truck in front. A tank engine is one which carries the feed water in tanks on the engine itself instead of in the tender, as in other engines. The various different forms are too numerous even to name.

There has been some effort made to introduce compounding in locomotive practice. It has in some cases been very successful, especially for express trains. A committee of the American Railway Master Mechanics Association says of compounding:

"(a) It has achieved a saving in the fuel burnt, averaging 18 per cent at reasonable boiler pressures.

- (b) It has lessened the amount of water to be handle.
- (c) The tender can, therefore, be reduced in size and weight.
- (d) It has increased the possibilities of speed beyond sixty miles per hour, without unduly straining the engine.
- (e) It has increased the haulage power at full speed.
- (f) In some classes of engines it has increased the starting power.
- (g) It has lessened the valve friction per horse-power developed."

A number of other reasons are given in their report.

In opposition to this may be mentioned the extra first cost of the engine and the cost of maintaining a more complicated machine. It is much more work to keep it in repair and many engineers are of the opinion that the saving in fuel is only sufficient to offset these extra expenses. If the engine is running under steady load, the compounding will effect a great saving; but in many parts of the country the load varies constantly, due to grades in the road.

We shall learn later that a compound engine cannot work efficiently under light load. If the grades are first up and then down, the simple engine is the more economical. For a steady up-grade the compound is more economical, as it can be run steadily under full load. This is especially true in mountain districts, where the long up-grades and scarcity of fuel and water make ideal conditions for compound locomotives. Through freight service probably offers the widest field for success with these engines.

It has already been said that it is difficult to balance an engine completely. This defect is very much greater in locomotives than in stationary engines. Lack of balance in a locomotive results in serious pounding of the track. Also there is danger of flattening and breaking the wheels, and the rails may be seriously damaged.

Pumping Engines. The first steam engines built were pumping engines and today the most economical engines are those built for this work.

In pumping engines it is not absolutely necessary to have a revolving shaft. All that is required is that the piston in the pump cylinder shall be driven back and forth with a plain recipro-

ating motion which may be exactly like that of the steam piston. For this reason, in early pumping engines and also in many modern engines, the reciprocating motion of the steam piston is applied directly, or through a beam, to produce the reciprocating motion of the pump piston or plunger without the use of any revolving part. Frequently, however, it is desirable to use a fly wheel so that the steam may be used expansively, and in these cases, of course, a revolving shaft must be used. Fig. 19 shows a power pump.

For deep-well or mine pumping, the cylinders are often set in a vertical position directly over the pump cylinder. The piston rod extends from the steam cylinder directly below to the pump plunger. Sometimes it is possible to use steam expansively in these pumps by reason of the weight of the reciprocating parts. When the weight is sufficient, the steam can be cut off before the end of

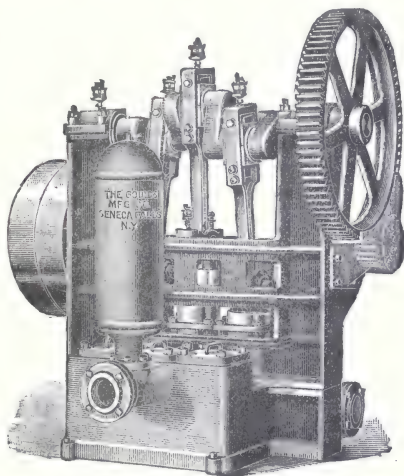


Fig 19.

the stroke and the momentum of the parts will be enough to just finish the stroke, consequently these pumps are sometimes compounded. They are possible only in pumping from very deep wells.

Direct-acting Steam Pumps. Fly wheel pumps have one disadvantage, if run too slowly the momentum of the fly wheel is not sufficient to carry it by the dead centers; if run too fast the fly wheel is in danger of bursting. A fly-wheel pump can be made to discharge a small amount of water by means of a bypass valve, but of course it then runs at a disadvantage.

The direct-acting steam pump shown in Fig. 20 has the steam piston at one end of a rod and the water piston at the other end. The steam pressure acts directly on the piston; no fly wheel is used, and since the reciprocating parts are comparatively light, and there is no revolving mass to carry by the dead points, it is evident that in the ordinary form there can be no expansion of steam. The pump is inexpensive and gives a positive action. It uses a great quantity of steam relatively, but for small work the absolute amount is not very great. Even in larger engines the lighter foundations that are possible and the slight first cost are frequently controlling features.

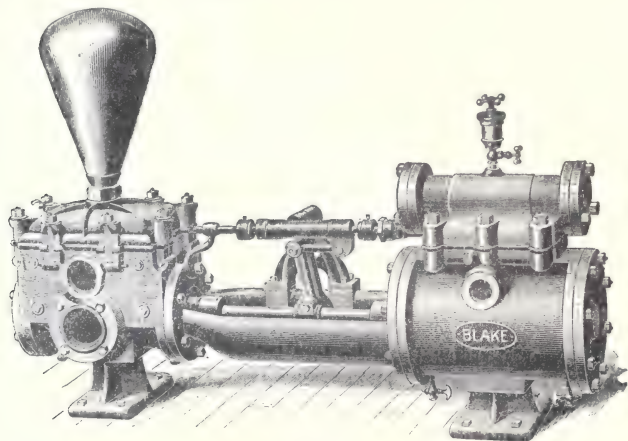


Fig. 20.

A rocker or bell-crank lever on the piston rod moves the steam valve and admits steam to the other side of the piston while opening the first side to exhaust. In large pumps of this kind, and even in some small ones, this motion merely admits steam to a small auxiliary piston which then moves the main steam valve by steam pressure. Some pumps operate the steam valve by means of a tappet instead of a rocker and bell-crank lever.

There have been various devices tried for using steam expansively in these direct-acting pumps without the use of a fly-wheel.

In order to do this it is necessary to provide some means of storing energy during the early part of the stroke and returning it during the latter part, when, owing to the expansion, the pressure of the steam is less. One such device is as follows: a crosshead A, (Fig. 21) fixed to the piston rod is connected to the plungers of a pair of oscillating cylinders B B, which contain water and communicate with a reservoir full of air compressed to about 300 pounds per square inch. When the stroke (which takes place in the direction of the arrow) begins, these plungers are first forced in, and hence work is at first done on the main piston rod, through the compensating cylinders B B, on the compressed air in the reservoir. This continues until the crosshead has advanced so that the oscillating cylinders stand at right angles to the line of stroke. Then for the remainder of the stroke their plungers

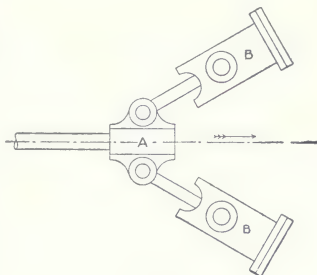


Fig. 21.

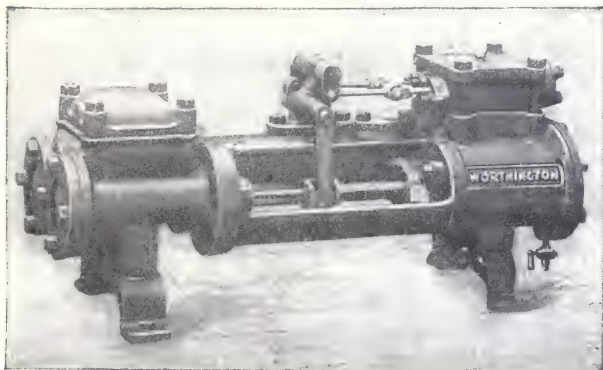


Fig. 22.

assist in driving the main piston, and the compressed air gives out the energy which it stored in the earlier portion.

The *Duplex steam pump* consists simply of two direct-acting steam pumps placed side by side, as shown in Fig. 22. On the piston rod on one side is a bell-crank lever which operates the valve of the other pump. On the further piston rod is a rocker arm which moves the valves of the first pump. There must be a rocker on one side and a bell-crank lever on the other because of the relative motion of the valves and pistons. The first piston, as it goes forward, must use a rocker because it draws the second valve back. The second piston, as it goes back, must use a bell-crank lever because it must push the first valve back in the same direction as its own motion. The two pistons are made to work a half-stroke apart. Thus one begins its stroke when the other is in the

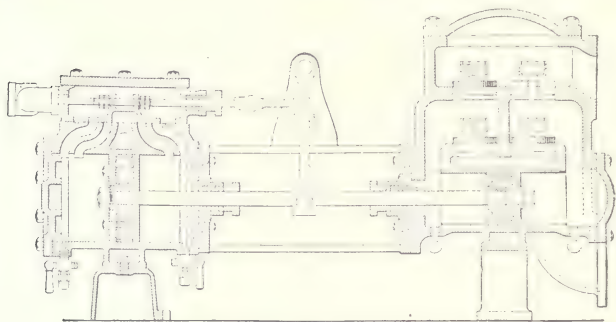


Fig. 23.

middle. In this way a steadier flow of water is obtained, for both pumps discharge into the same delivery pipe. The pumps may be made compound. A sectional view of the pump is shown in Fig. 23.

Corliss Engines. In large engines a common way of regulating the steam supply is by changing automatically the point in the stroke of the piston at which the steam is cut off. This is frequently accomplished by using some trip gear similar to the one first introduced by Geo. Corliss. These gears are called Corliss gears. In the Corliss gear there is a separate admission valve and a separate exhaust valve for each end of the cylinder, as shown in Fig. 24. The exhaust valves are opened and closed by

the motion of rods or cranks connected to them. The admission valves are opened in the same way, but they snap shut by themselves when the piston has reached a certain point of its stroke. This point will vary with the position of the governor, which in turn depends on the speed of the engine. These Corliss engines cannot be run at high speed because the trip gear requires some time to act.

The valves of Corliss engines turn in hollow cylindrical seats which extend across the cylinders. A wrist plate which turns on

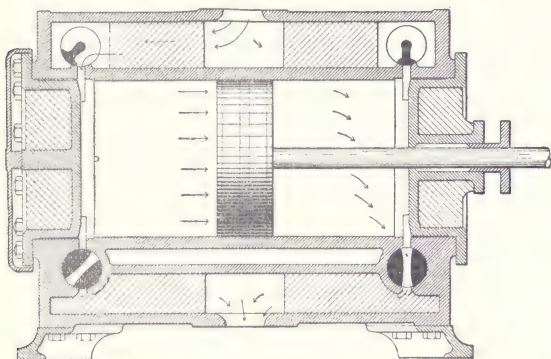


Fig. 24.

a pin on the outside of the cylinder receives a motion of oscillation from an eccentric and opens the valves by means of the rod connections. When the piston has reached a point where the steam should be cut off, the trip gear is held in such a position by the governor that it releases the valve, which now springs shut under the action of the dash-pot. The admission valve to the other side of the cylinder is controlled in exactly the same way.

The admission valves are generally placed at the top and the exhaust valves at the bottom of the cylinder. Any water which may be formed by steam condensing can readily drain off by this arrangement. There are a great many modifications of the Corliss gear. Fig. 25 is a Harris Corliss Engine.

The advantage of the Corliss gear is the great range through

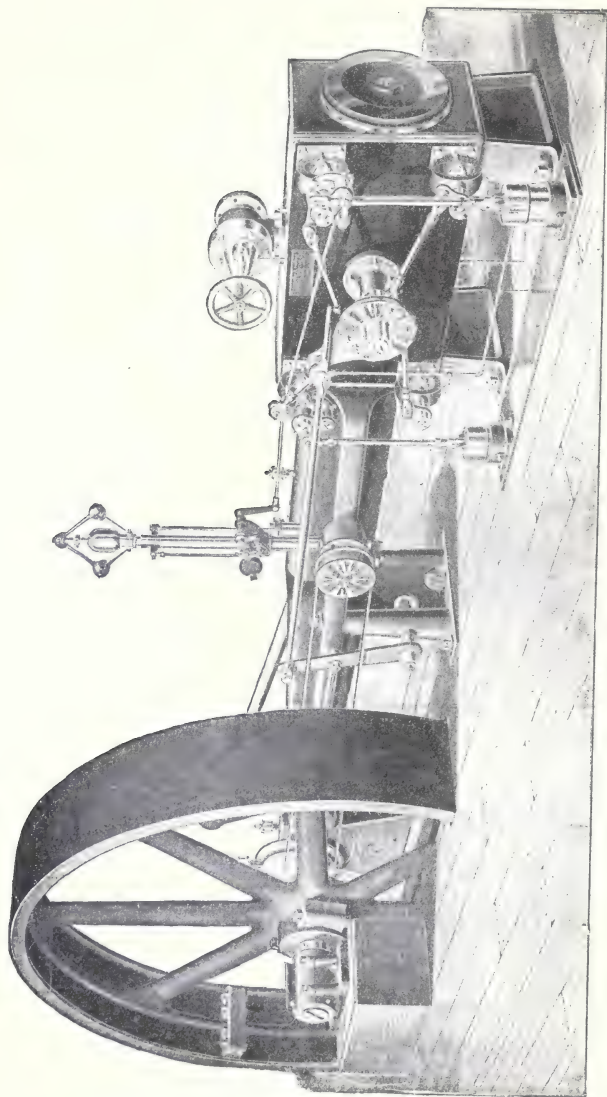


Fig. 25.

which the cut-off can be varied, from very early to very late in the stroke. Another great advantage is the quick action which reduces wire drawing. To understand fully the loss caused by wire drawing requires a knowledge of higher mathematics.

CONDENSERS.

When low-pressure steam is cooled it gives up its latent heat; that is, it changes from a vapor to a liquid. We know that a liquid occupies much less space than an equal weight of vapor; hence, by changing the steam to water the pressure is greatly reduced. By cooling the steam in the cylinder in front of the piston the back pressure, or resistance, is decreased. This reduces the pressure necessary to push the piston through the stroke, therefore less steam is required to do the work. This cooling is accomplished by some form of condenser.

There are two types of condensers, surface and jet. A surface condenser is one in which the steam passes through pipes surrounded by water or the water flows through pipes surrounded by steam. In the jet condenser the steam is condensed by coming in contact with cold water, which enters as spray. In both types the steam is condensed to water and a partial vacuum is formed, because water occupies much less space than does an equal weight of steam. If it were not for the air in the entering steam there would be an almost perfect vacuum. For this reason every condenser is fitted with an air pump to remove this air and the condensed steam.

Surface Condensers. The condenser shown in section in Fig. 26 is a common form of the surface type, in which the air pump and circulating pump are both direct acting and are operated by the same steam cylinder. The cold condensing water is drawn from the supply into the circulating or water pump. This pump forces the water up through the valves and water inlet to the condenser. It flows, as indicated by the arrows, through the inner tubes of the lower section, then back through the space between the inner and outer tubes. The water then passes upward and through the upper section, as it did in the lower. It then passes out of the condenser through the water outlet, taking with it the heat it has received from the steam.

The exhaust steam from the engine enters at the exhaust inlet and comes in contact with the perforated plate, which scatters it among the tubes. This method protects the upper tubing from the effect of direct contact with the exhaust steam. The steam expanding in the condenser comes in contact with the cool tubes,

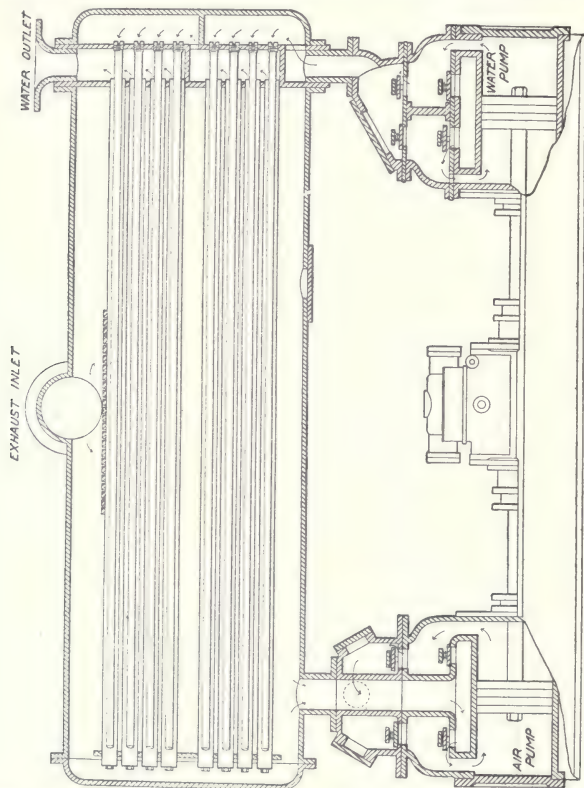
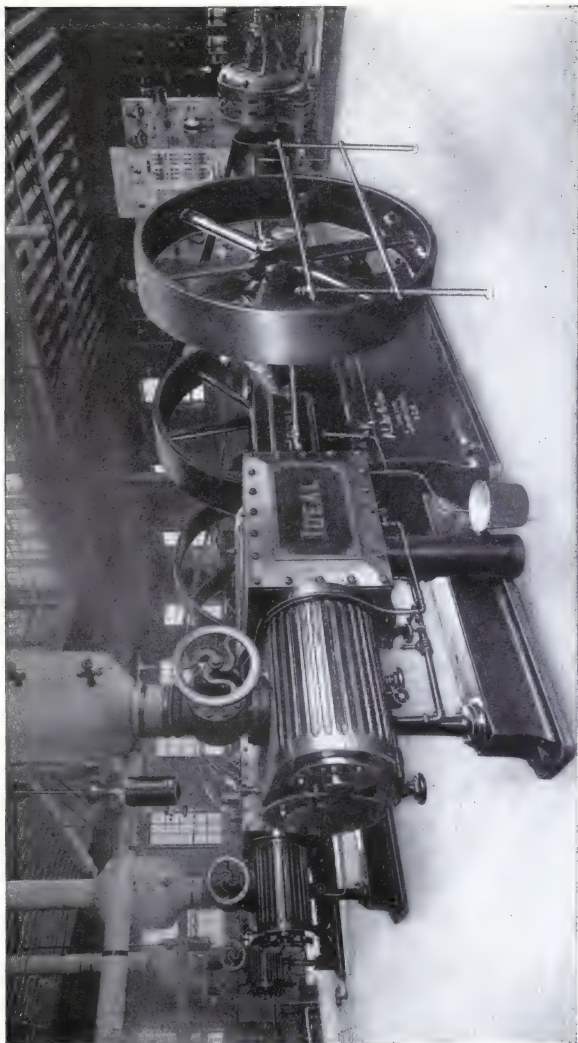


Fig. 26.

through which cold water is circulating, and condenses. The air pump draws the air and condensed steam out of the condenser and thus maintains a partial vacuum. This causes the exhaust steam in the engine cylinder to be drawn into the condenser.



ENGINE ROOM, POSTUM CEREAL CO., SHOWING THREE IDEAL TANDEM-COMPOUND ENGINES, BELT-CONNECTED TO GENERATORS
A. L. Ide & Sons, Springfield, Ill.

The condensed exhaust steam collects at the bottom of the condenser, is drawn into the air pump cylinder and is discharged while heated to the hot well of the boiler. The use of this hot water as feed water is a considerable saving; but the great advantage of the condenser is the reduction in back pressure.

Hot water cannot be used by an ordinary pump as well as cold water because of the pressure of the vapor which arises from the hot water. In the condenser shown, the water and air pumps are run by the piston in the steam cylinder. Sometimes these pumps are connected to the main engine and receive motion from the shaft or crosshead.

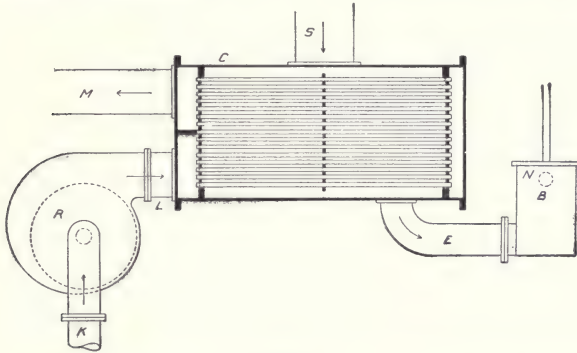


Fig. 27.

The general arrangement of the surface condenser with the necessary pumps is shown in Fig. 27. The cooling water enters through the pipe K, and flows to the circulating pump R, which forces the water into the condenser through the pipe L. In case the water enters the condenser under pressure from city mains no circulating pump is necessary. After flowing through the tubes it leaves the condenser by means of the exit M, and flows away. Exhaust steam enters at S, and is condensed by coming in contact with the cold tubes; the water (condensed steam) then falls to the bottom of the condenser and flows to the air pump B by the pipe E. The air pump removes the air, vapor and condensed steam from the condenser and forces it through the pipe N into the hot well, from which it goes to the boilers or to the feed tank.

The circulating pump, when separate from the condenser, is usually of the centrifugal type. This pump consists of a fan or wheel which is made up of a central web or hub, and arms or vanes. This pump is shown in Fig. 28. The vanes are curved, and as the water is drawn in at the central part the vanes throw it off at the circumference. A suitable casing directs the flow. This type of pump is advantageous because there are no valves to get out of order, and as the lift is little, if any, the pump will discharge a large volume of water in a nearly constant stream. The circulating pump is usually so placed that the water flows to it under a slight head. The pump is driven by an independent engine so that the circulating water may cool the condenser even if the main engine is not working.

The centrifugal pump works more smoothly and with less trouble than an ordinary force pump, because it is not reciprocating and it has no valves.

Jet Condensers. Fig. 29 shows the longitudinal section of an independent jet condenser with the pump. The cold water used to condense the steam enters at A, passes down the spray pipe B, and is broken into a fine spray by means of the spray cone C. This action insures a rapid and thorough mixing of the steam and water and consequently a rapid condensation. The exhaust steam enters at D, with a comparatively high velocity, which is imparted to the water. The whole mixture of water, steam and vapor passes at high velocity through the conical chamber E to the pump cylinder F. The pump forces this water to the pipe G. The spray cone is adjusted by the stem which passes through the stuffing box at the top of the condenser. The valves are shown at H and K. The steam end of the pump is at L. Motion of the valve is produced by the rocker arm J.

In Fig. 30 a jet condenser is shown connected to a stationary engine and boiler. The exhaust pipe leads from the engine to the

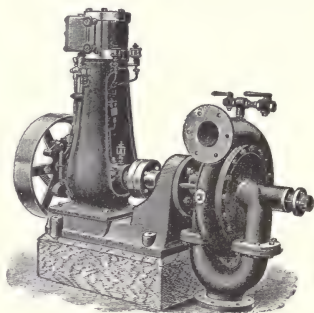


Fig. 28.

condenser, the arrows indicating the direction of flow. Cold water enters the condenser through a pipe connected to the well. Part of the mixture of exhaust steam and condensed water goes to the feed-water heater, which is kept nearly full; the rest passes to the sewer. The heater is placed a little above the feed pump, so that the water will enter the pump under a slight head, because the pump cannot raise water warmed by exhaust steam as readily as cold water.

The surface condenser is used almost universally in marine practice. Its first cost is more than that of the jet condenser and it requires more condensing water, but it allows only the condensed steam to return to the boiler. It is also used in stationary work when the condensing water is very impure. The jet condenser is not adapted for marine work, as it pumps both the condensing water and the condensed steam to the hot well. Hence, if salt water or water containing lime is used, it will enter the boiler and form sediment and scale. This type is used

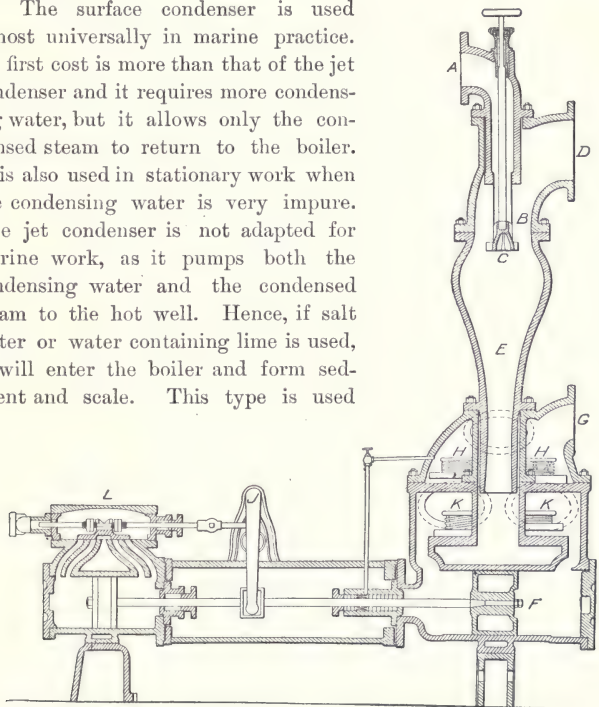


Fig. 20.

where fresh and moderately pure water is available.

It has been mentioned before that Watt always condensed the exhaust steam from his engines, and that when higher pres-

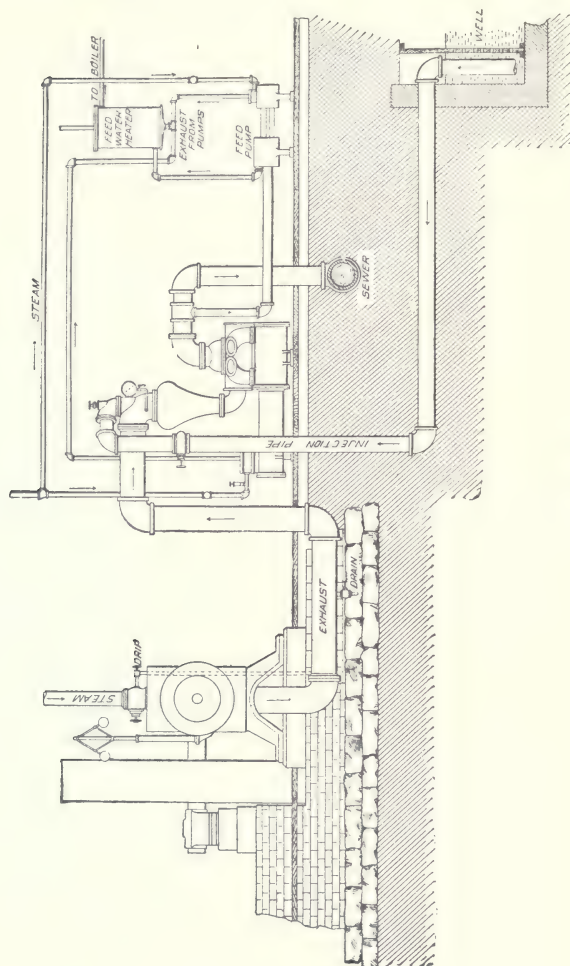


Fig. 30.

tures came into use some makers let the steam discharge into the atmosphere. This leads to the distinction between condensing and noncondensing engines. Both types are in common use, but the condensing engines are much more economical than the noncondensing, as far as fuel is concerned; but to condense the steam, considerable water is necessary, and condensing engines cost more and require more care. Consequently in some cases it is quite as economical, all things considered, to use the noncondensing engine.

COOLING TOWERS.

It sometimes happens that it is impossible to place a steam plant in close proximity to a natural water supply. In such cases the water necessary for the condenser (the circulating water) is expensive, and if the cost is very great it does not pay to add the condenser, because the cost of the circulating water might more than offset the gain from condensing. If, however, some means could be provided whereby the circulating water as it issues from the condenser could be cooled and then used over again in the condenser, the noncondensing engine could then be run condensing; thus taking advantage of all the benefits due to the use of reduced back pressure and heating of the feed water. This has been attempted by conducting the heated discharge water to a pond, where it is allowed to cool to a lower temperature before being used again. Another plan is to place in the yard or on the roof of the building large shallow pans, in which the water is cooled by being exposed to the atmosphere. These methods are unsatisfactory on account of the considerable area necessary and the slow action. In addition they are uncertain, because they are dependent upon atmospheric conditions.

A more efficient and at the same time more expensive process is to use a cooling tower or a water table. Fig. 31 shows the general arrangement of a cooling tower located upon the roof of a building. The discharge from the condenser is led, as shown by the arrows, to the top of the cooling tower, where it is cooled before being returned to the condenser. This cooling is effected by distributing the water, by a system of piping, to the upper edge of a series of mats or slats, over the surface of which the water

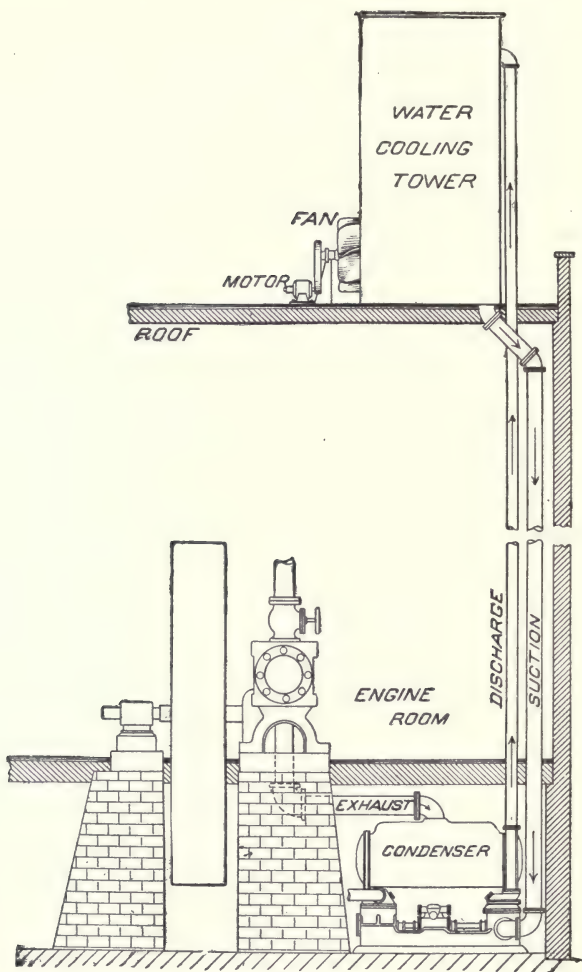


Fig. 31.

flows in a thin film to a reservoir which is situated in the bottom of the cooling tower. The mats partially interrupt the flow, and by breaking up the water in small streams cause new portions to be exposed to the cooling effect of the air currents. The water from the reservoir then flows downward through the suction pipe, and is pumped by the circulating pump through the condenser. After passing through the condenser and absorbing heat from the exhaust steam, it rises through the discharge pipe and commences the circuit over again.

The tower may have several arrangements and be made of various materials. A satisfactory form is constructed of steel plates; within the tower are a large number of mats of steel wire cloth galvanized after weaving.

To assist in the cooling of the water, the air is often made to circulate rapidly by means of a fan, which forces the air into the lower part of the tower and upwards among the mats. This fan is usually of the ordinary type, and may be driven by an electric motor, a line of shafting, or by a small independent engine.

In case the fan is not used, the mats are arranged so that they are exposed to the atmosphere, as shown in Fig. 32. This of course necessitates the removal of the steel casing. Usually the fanless tower must be placed at the top of a high building, or in some position where the currents of air can readily circulate among the mats.

With an efficient type of cooling tower the water may be reduced from 30 to 50°, thus allowing a vacuum of from 22 to 26 inches. This will of course greatly increase the economy of the plant, and allow the heated feed water to be returned to the boiler.

The water table is usually made of wooden slats placed in the ground near the plant. After trickling over the slats and becoming cooled by the air, it collects in the bottom of the reservoir and is then pumped into the condenser.

THE FLY WHEEL.

It is evident that while the piston can push the crank around during part of the stroke, and pull it during another part, there are still two places (called dead points) at the ends of the stroke, where the pressure on the piston, no matter how great, can exert

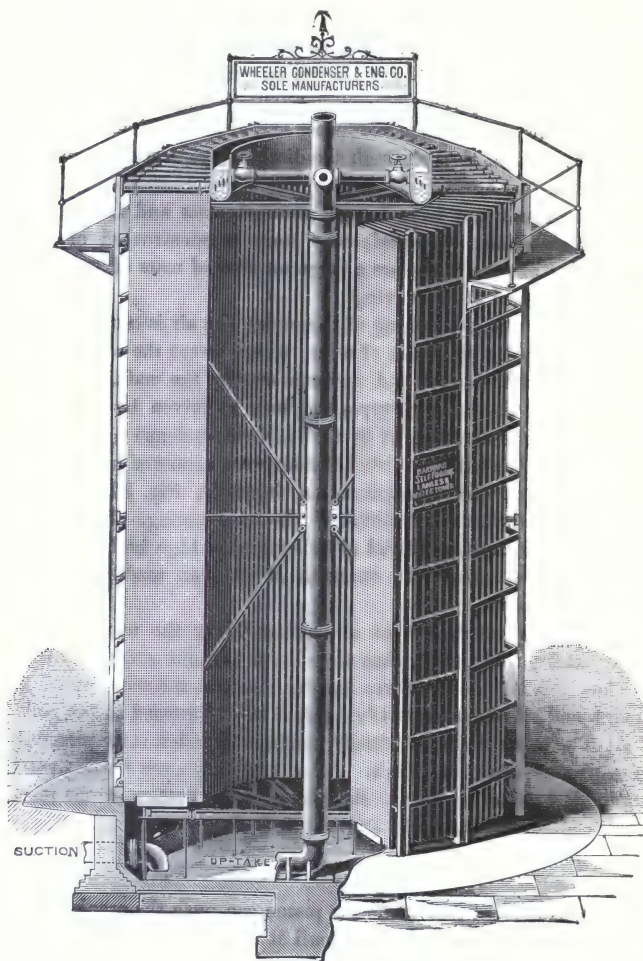


Fig. 32

no turning moment on the shaft. Therefore, if some means is not provided for making the shaft turn past these points without the assistance of the piston, it may stop. This means is provided in the fly wheel, which is merely a heavy wheel placed on the shaft. On account of the momentum of the fly wheel it cannot be stopped quickly, and therefore carries the shaft around until the piston can again either push or pull.

If a long period be considered, the mean effort and the mean resistance must be equal; but during this period there are temporary changes of effort, the excesses causing increase of speed. To moderate these fluctuations several methods are employed.

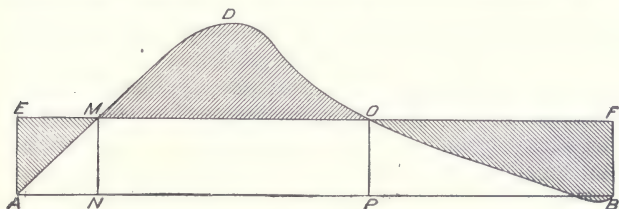


Fig. 33.

The turning moment on the shaft of a single cylinder engine varies, first, because of the change in steam pressure, and second, on account of the angularity of the connecting rod. Before the piston reaches mid-stroke the turning moment is a maximum, as shown by the curve, (Fig. 33). While near the ends of the stroke (the dead points) the turning moment diminishes and finally becomes zero. This, of course, tends to cause a corresponding change in the speed of rotation of the shaft. In order to have this speed as nearly constant as possible, and to give a greater uniformity of driving power, the engine may be run at high speed. By this means the inertia of the revolving parts, such as the connecting rod and crank, causes less variation. When the work to be done is steady and always in the same direction (as in most factories), a heavy fly wheel may be used.

The heavier the fly wheel, the steadier will be the motion. It is, of course, desirable in all factory engines to have steady motion, but in some it is more important than in others. For

instance, in a cotton mill it is absolutely necessary that the machinery shall move with almost perfect steadiness; consequently mill engines always have very large, heavy fly wheels. It is undesirable to use larger wheels than are absolutely necessary, because of the cost of the metal, the weight on the bearings and the danger from bursting.

If the turning moment which is exerted on the shaft from the piston could be made more regular, and if dead points could be avoided, it would be possible to get a steadier motion with a much smaller fly wheel.

If the engine must be stopped and reversed frequently, it is better to use two or more cylinders connected to the same shaft. The cranks are placed at such angles that when one is exerting its minimum rotative effort, the other is exerting its maximum turning moment; or, when one is at a dead center, the other is exerting its greatest power. These two cylinders may be identically the same, as is the case with most hoisting engines and with many locomotives; or the engine may be compound or triple expansion.

This arrangement is also used on engines, for mines, collieries, and for hoisting of any sort where ease of stopping, starting and reversing are necessary. Simple expansion engines with their cranks at right angles are said to be coupled.

The governor adjusts the power of the engine to any large variation of the resistance. The fly wheel has a duty to perform which is similar to that of the governor. It is designed to adjust the effort of the engine to sudden changes of the load which may occur during a single stroke. It also equalizes the variation in rotative effort on the crank pin. The fly wheel absorbs energy while the turning moment is in excess of the resistance and restores it while the crank is at or near the dead points. During these periods the resistance is in excess of the power.

The action of the fly wheel may be represented as in Figs. 33 and 34. It will be noticed that in Fig. 33 the curve of crank effort runs below the axis toward the end of the stroke. This is because the compression is greater than the pressure near the end of expansion, and produces a resultant pressure on the piston. In Fig. 34 the effect of compression has been neglected. Let us suppose that the resistance, or load, is uniform. In Fig. 33 the line

A B is the length of the semi-circumference of the crank pin, or it is the distance the crank pin moves during one stroke. The curve A M D O B is the curve of turning moment for one stroke. M N is the mean ordinate, and therefore A E F B represents the constant resistance. The effort and resistance must be equal if the speed is uniform; hence $A E F B = A M D O B$. Then $\text{area } A E M + \text{area } O F B = \text{area } M D O$. At A the rotative effort is zero because the crank pin is at the dead point; from A to N the turning moment is less than the resistance. At N the resistance and the effort are equal. From N to P the effort is in excess of the resistance. At P the effort and resistance are again equal. From P to B the resistance is greater than the effort. In other words, from A to N the work done by the steam is less than the resist-

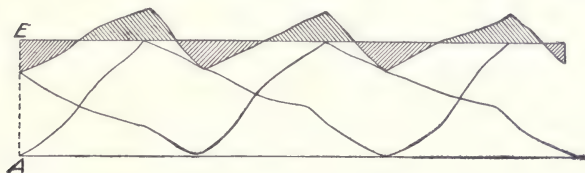


Fig. 34.

ance. This shows that the work represented by the area A E M must have been done by the moving parts of the engine. From N to P the work done by the steam is greater than the resistance, and the excess of energy is absorbed by or stored in the moving parts. From P to the end of the stroke, the work represented by the area O F B is done on the crank pin by the moving parts.

We know from the formula, $E = \frac{W V^2}{2g}$, that energy is proportional to the square of the velocity. Hence as W and g remain the same, the velocity must be reduced when the moving parts are giving out energy, and increased when receiving energy. Thus we see that the tendency of the crank pin is to move slowly, then more rapidly. The revolving parts of an engine have not sufficient weight to store this surplus energy, hence a heavy fly wheel is used.

In case there are two engines at right angles, two effort curves

must be drawn, as shown in Fig. 34. The mean ordinate A E is equal to the mean or constant resistance. There are two minimum and two maximum velocities in one stroke. The diagram shows that the variation is much less than for a single cylinder; hence a lighter fly wheel may be used.

The *weight of the fly wheel* depends upon the character of the work done by the engine. For pumping engines and ordinary machine work the effort need not be as constant as for electric lighting and fine work. In determining the weight of a fly wheel the diameter of the wheel must be known, or the ratio of the diameter of the wheel to the length of stroke. If the wheel is too large, the high linear velocity of the rim will cause too great a centrifugal force and the wheel is likely to break. In practice, about 6,000 feet per minute is taken as the maximum linear velocity of cast-iron wheels. When made of wood and carefully put together, the velocity may be taken as 7,000 to 7,500 feet per minute.

We know that linear velocity is expressed in feet per minute by the formula, $V = 2\pi R N$, or $V = \pi D N$.

Then if a wheel runs at 100 revolutions per minute, the allowable diameter would be,

$$6,000 = 3.1416 \times D \times 100$$

$$D = \frac{6,000}{3.1416 \times 100} = 19.1 \text{ feet.}$$

If a wheel is 12 feet in diameter the allowable speed is found to be,

$$N = \frac{V}{\pi D}$$

$$= \frac{6,000}{3.1416 \times 12} = 159 \text{ revolutions per minute.}$$

It is usual to make the diameter a little less than the calculated diameter.

Having determined the diameter, the weight may be calculated by several methods. There are many formulas to obtain this result given by various authorities. One formula is given as follows:

$$W = \frac{C \times d^2 \times b}{D^2 \times N^2}$$

In the above, W = weight of the rim in pounds
 d = diameter of the cylinder in inches
 b = length of stroke in inches
 D = diameter of fly wheel in feet
 N = number of revolutions per minute

C is a constant which varies for different types and conditions.

Slide-valve engines, ordinary work, $C = 350,000$
 Corliss engines, ordinary work, $C = 700,000$
 Slide-valve engines, electric lighting, $C = 700,000$
 Corliss engines, electric lighting, $C = 1,000,000$
 Automatic high-speed engines, $C = 1,000,000$

Example. Let us find the weight of a fly-wheel rim for an automatic high-speed engine used for electric lighting. The cylinder is 24 inches in diameter; the stroke is 2 feet. It runs at 300 revolutions per minute, and the fly wheel is to be 6 feet in diameter.

$$W = 1,000,000 \times \frac{(24)^2 \times 24}{36 \times 90,000}$$

$$W = 4,266 \text{ pounds}$$

Another example. A plain slide-valve engine for electric lighting is 20" $\times$ 24". It runs at 150 revolutions per minute. The fly wheel is to be 8 feet in diameter. What is the weight of its rim?

$$W = 700,000 \times \frac{400 \times 24}{64 \times 22,500}$$

$$W = 4,666 \text{ pounds.}$$

The weight of a fly wheel is considered as being in the rim. The weight of the hub and arms is simply extra weight. Then, if we know the weight of the rim and its diameter, we can find the width of face and thickness of rim. Let us assume the given diameter to be the mean of the diameter of the inside and outside of the rim.

Let b = width of face in inches

t = thickness of rim in inches

d = diameter of fly wheel in inches

.2607 = weight of 1 cubic inch of cast iron

Then,

$$\begin{aligned} W &= .2607 \times b \times t \times \pi d \\ &= b \times t \times .819 d \end{aligned}$$

Suppose the rim of a fly wheel weighs 6,000 pounds, is 9 feet in diameter, and the width of the face is 24 inches. What is the thickness of the rim?

$$\begin{aligned} t &= \frac{W}{.819 \, d \, b} \\ &= \frac{6,000}{.819 \times 108 \times 24} \\ &= 2.83 \text{ inches} \end{aligned}$$

In this case the rim would probably be made $2\frac{1}{8}$ inches thick. The total weight, including hub and arms, would probably be about 8,000 pounds.

GOVERNORS.

The load on an engine is never constant, although there are cases where it is nearly uniform. While the engine is running at constant speed, the resistance at the fly-wheel rim is equal to the work done by the steam. If the load on the engine is wholly or partially removed, and the supply of steam continues undiminished, the force exerted by the steam will be in excess of the resistance. Work is equal to force multiplied by distance; hence, with constant effort, if the resistance is diminished, the distance must be increased. In other words, the speed of the engine will be increased. The engine will "race," as it is called. Also, if the load increases and the steam supply remains constant, the engine will "slow down."

It is evident, then, that if the speed is to be kept constant some means must be provided so that the steam supply shall at all times be exactly proportional to the load. This is accomplished by means of a governor.

Steam-engine governors act in one of two ways: they may regulate the *pressure* of steam admitted to the steam chest, or they may adjust the speed by altering the *amount* of steam admitted. Those which act in the first way are called throttling governors, because they throttle the steam in the main steam pipe. Those of the latter class are called automatic cut-off governors, since they automatically regulate the *cut-off*.

Theoretically, the method of governing by throttling the

steam causes a loss in efficiency, but the throttling superheats the steam, thus reducing cylinder condensation. By the second method the loss in efficiency is very slight, unless the ratio of expansion is already great, in which case shortening the cut-off causes more cylinder condensation. This subject will be taken up in detail later.

In most governors, centrifugal force, counteracted by some other force, is employed. A pair of heavy masses (usually iron balls or weights) are made to revolve about a spindle, which is driven by the engine. When the speed increases, centrifugal force increases, and the balls tend to fly outward; that is, they revolve in a larger circle. The controlling force, which is usually gravity or springs, is no longer able to keep the balls in their former path. When, therefore, the increase is sufficiently great, the balls in moving outward act on the regulator, which may throttle the steam or cause cut-off to occur earlier.

With the throttling governor a balanced throttle valve is placed in the main steam pipe leading to the valve chest. If the engine runs faster than the desired speed, the balls are forced to revolve at a higher speed. The increase in centrifugal force will cause them to revolve in a larger circle and in a *higher plane*. By means of some mechanism (levers and gears) the spindle may be forced downward, thus partially closing the valve. The engine, therefore, takes steam at a lower pressure, and the power supplied being less, the speed falls slightly.

Similarly, if the load is increased, the engine slows down, which causes the balls to drop and open the valve more widely, steam at higher pressure is then admitted, and the speed is increased to the regular number of revolutions.

With the Corliss or Wheelock engine the governor of this type acts differently. Instead of throttling the steam in the steam pipe, the governor is connected to the releasing gear by rods. An increase of speed causes the releasing gear to unhook the disengaging link earlier in the stroke. This causes earlier cut-off, which of course decreases the power and the speed, since the amount of steam admitted is less. If for any reason the load increases, the governor causes the valves to be held open longer. The cut-off, therefore, occurs later in the stroke.

One of the most common forms of governor is similar to that invented by James Watt. It is called, from its appearance, the pendulum governor. It is shown in Fig. 35. To consider the theory of the pendulum governor, the masses of the balls are assumed to be concentrated at their centers, and the rods made of some material having no weight.

When the governor is revolving about its axis at a constant speed the balls revolve in a circle having a radius r . The distance from this plane to the intersection of the rods, or the rods produced, is called the height and is equal to h .

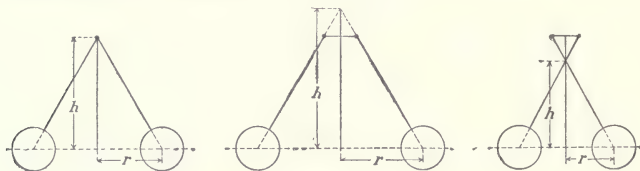


Fig. 35.

If the balls revolve faster, the centrifugal force increases, r becomes greater and h diminishes. We know that centrifugal force is expressed by the formula,

$$F = \frac{Wv^2}{g r}$$

Then centrifugal force varies inversely as the radius.

While the pendulum is revolving, centrifugal force acts horizontally outward and tends to make the balls fly from the center; gravity tends to make the balls drop downward. In order that the balls shall revolve at a certain height, the moments of these two forces about the center must be equal, or the weight of the balls multiplied by their distance from the center must equal the centrifugal force multiplied by the height, or,

$$W \times r = F \times h$$

from which,

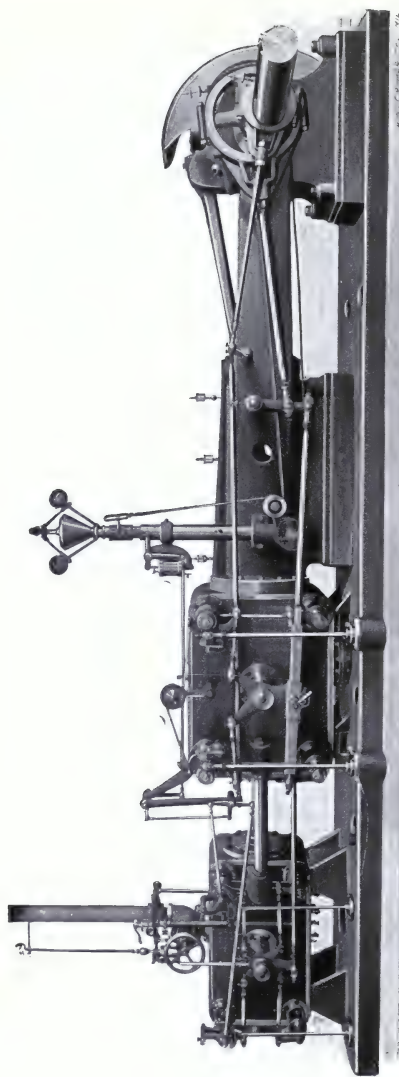
$$\frac{h}{r} = \frac{W}{F}$$

or,

$$\frac{h}{r} = \frac{W}{\frac{Wv^2}{g r}} = \frac{g r}{v^2}$$

from which,

$$h = \frac{g r^2}{v^2}$$



TANDEM CORLISS ENGINE DRIVING ELECTRIC LIGHT PLANT. TRADESMEN'S NATIONAL BANK, PITTSBURGH.
Nordberg Manufacturing Company.

Now we know that the linear velocity of a point revolving in the circumference of a circle is expressed as $2 \pi r N$ feet per second.

$$\text{Then, } h = \frac{g r^2}{v^2} = \frac{g r^2}{4 \pi^2 r^2 N^2} = \frac{g}{4 \pi^2 N^2}$$

Since we know the values of g and π we can write the formula,

$$h = \frac{32.16}{4 \times 3.1416^2 \times N^2} = \frac{.8146}{N^2} \text{ feet per second, or}$$

$$h = \frac{9.775}{N^2} \text{ inches per second.}$$

If N is the number of revolutions per minute, since $60^2 = 3600$,

$$h = \frac{2,932.56}{N^2} \text{ feet}$$

$$h = \frac{35,190.7}{N^2} \text{ inches}$$

From the above formula, we see that the height is independent of the weight of the balls or the length of the rod; it depends upon the number of revolutions. The height varies inversely as the square of the number of revolutions.

The ordinary pendulum governor is not isochronous; that is, it does not revolve at a uniform speed in all positions; the speed changes as the angle between the arms and the spindle changes.

The early form consisted of two heavy balls suspended by links from a pin connection in a vertical spindle, as shown in Figs. 36 and 37. The spindle is caused to revolve by belting or gearing from the main shaft, so that as the speed increases, centrifugal force causes the balls to revolve in a circle of larger and larger diameter. The change of position of these balls can be made to affect the controlling valves so that the admission or the throttling shall vary with their position. With this governor it is evident that for a given speed of the engine there is but one position possible for the governor; consequently one amount of throttling or one point of cut-off, as the case may be. If the load varies, the speed of the engine will change. This causes the position of the governor balls to be changed slightly, thus altering the pressure.

But in order that the pressure or cut-off shall remain changed, the governor balls must stay in their new position. That is to say, the speed of the engine must be slightly changed. Thus with the old ball governors there was a slightly different speed for each load. This condition has been greatly improved by various modifications until now such governors give excellent regulation.

While the engine is running with a light load, the valve controlled by the governor will be open just enough to admit steam at a pressure that will keep the engine running at a given speed. Now, if the engine is heavily loaded, the throttle valve must be wide open. The change of opening is obtained by a variation in the height of the governor, which is caused by a change of speed. Thus we see that the governor can control the speed only within certain limits which are not far apart. The difference in the extreme heights of the governor must be sufficient to open the throttle its entire range. In most well-designed engines the speed will not vary more than 4 per cent; that is, 2 per cent above or below the mean speed.

From the formula $h = \frac{35,190.7}{N^2}$, we can compute the heights corresponding to given speeds as shown by the following table:

Number of Revolutions per Minute.	Height in Inches.	Variation of Height in Inches 4 per cent.
250	.563	.0225
200	.879	.035
175	1.149	.046
150	1.564	.062
125	2.252	.090
100	3.519	.140
75	6.256	.250
50	14.076	.563

In the above table the second column is found from the formula $h = \frac{35,190.7}{N^2}$. The third column is the variation in height for a speed variation of 4 per cent or 2 per cent either above or below the mean.

From the table we see that for a considerable variation of speed there is but slight variation in the height of the governor. Also for high speeds the height of the governor is so small that it would be difficult to construct it. The slight variation in height is too small to control the cut-off or throttling mechanism throughout the entire range.

Other disadvantages of the fly-ball governor are as follows: it is apparent that the valves must be controlled by the weight of the governor balls. In large engines this requires very heavy balls in order to quickly overcome the resistance of the valves. But these large balls have considerable inertia and will therefore be reluctant to change their speed with that of the engine. The

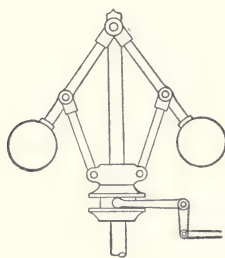


Fig. 36.

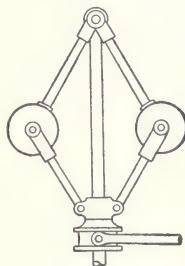


Fig. 37.

increased weight will also increase the friction in the governor joints, and the cramping action existing when the balls are driven by the spindle will increase this friction still further. All these things tend to delay the action of the governor, so that in all large engines the old-fashioned governor became sluggish. The balls had to turn slowly because they were so heavy; this was especially troublesome in high-speed engines.

To remedy these defects the weighted or Porter governor was designed. (See Fig. 38.) It has a greater height for a given speed, and the variation in height for a given variation of speed is greater. When a governor has this latter quality, that is, a great variation in height for a given variation of speed, it is said to be sensitive. By increasing this variation in height the sensitiveness is increased. Thus, if a governor running at 50 revolutions has a

variation in height of .57 inch, it is not as sensitive as one having a variation of 1 inch for the same speed.

In the weighted governor, the weight is formed so that the center of gravity is in the axis. It is placed on the spindle and is free to revolve. The weight adds to the weight of the balls, and thus increases the moment of the weight. It does not, however, add to the centrifugal force, and hence the moment of this force is unchanged. We may then say the weight adds effect to the weight but not to the centrifugal force, and as a consequence the height of the governor, for a given speed, is increased. If we let W equal the combined weight of the balls as before, and W' equal the added weight, the moments are,

$$(W + W') \times r = F h$$

$$(W + W') \times r = \frac{W v^2}{g r} \times h$$

$$\begin{aligned} h &= (W + W') r \times \frac{g r}{W v^2} \\ &= \frac{(W + W') r^2 g}{W \times 4 \pi^2 r^2 N^2} \\ &= \frac{(W + W')}{W} \times \frac{g}{4 \pi^2 N^2} \end{aligned}$$

$$\text{We know that } \frac{g}{4 \pi^2 N^2} = \frac{.8146}{N^2}.$$

$$\text{Then } h = \left\{ \frac{W + W'}{W} \right\} \times \frac{.8146}{N^2}.$$

Hence the height of a weighted governor is equal to the height of a simple pendulum governor multiplied by

$$\left\{ \frac{W + W'}{W} \right\} \text{ or } \left\{ 1 + \frac{W'}{W} \right\}.$$

For instance, if the height of a simple pendulum is 10 inches, and the weight of the balls equal to the added weight, the height will be,

$$\begin{aligned} h &= \left\{ \frac{1+1}{1} \right\} \times 10 \\ &= 2 \times 10 \end{aligned}$$

Thus we see that if a weight equal to the combined weight of the balls is added, the height of the governor will be doubled.

We know that if the balls fall, the cut-off comes later. If the belt driving the governor slips off or breaks, the balls will drop, and, making the cut-off later, will allow the engine to "run away." To diminish this danger many governors are provided with some kind of safety stop, which closes the valve when the governor loses its normal action. Usually a trip is provided which the governor does not touch in its normal positions, but which will be released if the balls drop down below a certain point.

In another arrangement, instead of a weight, a strong spring is used, and this makes it possible to put the governor in any position.

Spring Governors. In many cases a spring is used in place of the weight. This type of governor is used frequently on throttling engines; it consists of a pendulum governor with springs added to counteract the centrifugal force of the balls. Thus the height and sensitiveness are increased. Fig. 39 shows the exterior view of a Waters governor, and Fig. 40 the same governor having the safety stop. In this governor the weights are always in the same plane, the variation in height being due to the action of the bell crank levers connecting the balls and spindle. When the balls move outward the spindle moves downward and tends to close the valve. The governor balls revolve by means of a belt and bevel gears. The valve and seat are shown in section in Fig. 41. The valve is a hollow cylinder with three ports, by means of which steam enters the valve. The seat is made in four parts, that is, there are four edges that the steam passes as it enters the valve. The valve being cylindrical and having steam on both sides is balanced, and because of the many openings only a small travel is necessary.

Shaft Governors. Usually some form of pendulum governor

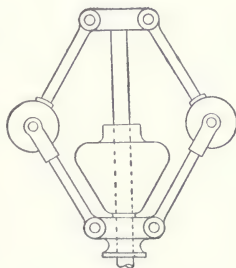


Fig. 38.

is used for throttling engines. For governing an engine by varying the point of cut-off, shaft governors are generally used; however, Corliss engines and some others use pendulum governors for this purpose. Cut-off governors are called shaft governors because they are placed on the main shaft; they are made in many forms, but the essential features of all are the same. Two pivoted masses or weights are arranged symmetrically on opposite sides of the shaft, and their tendency to fly outward when the speed increases is resisted by springs. The outward motion of the weights closes the admission valve earlier, and the inward motion closes it later. This change is effected by altering the position of the eccentric, either by changing the eccentricity or the angular advance.

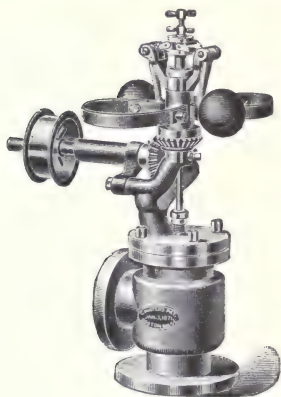


Fig. 39.

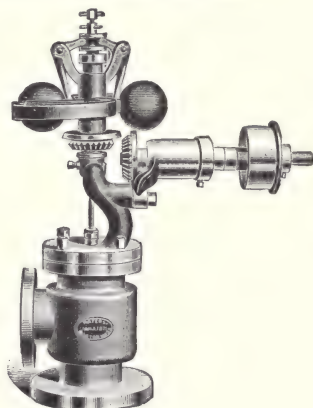


Fig. 40.

Shaft governors are made in a great variety of ways, no two types being exactly alike. If the principles of a few types are understood, it is easy to understand others. The following illustrates two common methods of shifting the eccentric.

Buckeye Engine Governor. The valve of the Buckeye engine is hollow and of the slide valve type. The cut-off valve is inside. The change of cut-off is due to the alteration of the angular advance. The arrangement of the parts which effect the change of angular advance is shown in Fig. 42. A wheel which

contains and supports the various parts of the governor is keyed to the shaft. Two arms, having weights *A A* at the ends, are pivoted to the arms of the wheel at *b b*. The ends having the weights are connected to the collar on the loose eccentric *C* by means of rods *B B*.

When the weights move to the position indicated by the dotted lines, the eccentric is turned on the shaft about a quarter of a revolution in the direction in which the engine runs. That is, the eccentric is advanced or the angular advance is increased. Now we know that if the angular advance is increased, cut-off occurs earlier. This is shown by the table on page 22 of "Valve Gears." If the engine had a single plain slide valve the variation of the angular advance would produce too great a variation of lead; but as this engine has a separate valve for cut-off, admission is not altered by the cut-off valve.

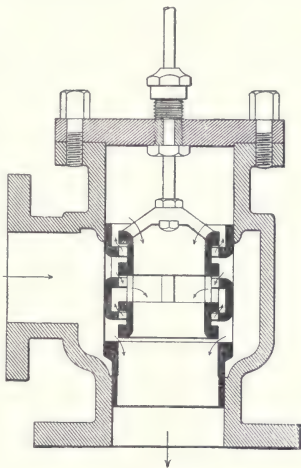


Fig. 41.

The springs *F F* balance the centrifugal force of the weights. The weights *A A* are varied to suit the speed; the tension on the springs is altered by means of the screws *c c*. Auxiliary springs are added in order to obtain the exactness of regulation necessary for electric lighting. These springs tend to throw the arms outward, but act only during the inner half of this movement.

The Straight-Line Engine Governor. Fig. 43 shows the governor of the straight-line engine. It has but one ball, *B*, which is linked to the spring *S* and to the plate *D E*, on which is the eccentric *C*. When the ball flies outward in the direction indicated by the arrow *F*, the eccentric is shifted about the pivot *O*; the links moving in the direction of arrow *H*. The ball is heavy and at a considerable distance from the center; hence it has a great centrifugal force, and the spring must be stiff.

The governor of the Buckeye engine alters the cut-off by changing the *angular advance*. The straight-line engine governor changes the *travel* of the valve. Shaft governors which alter the cut-off by changing the valve travel are very common.

LUBRICATION.

If two pieces of cast iron, just as they come from the foundry, are rubbed together, they will not slide over each other easily, because of little projections. If this same iron is filed or planed,

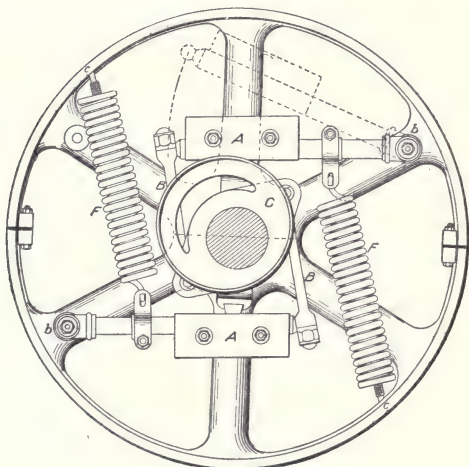


Fig. 42.

the pieces will slide much more easily. This is because the rough places have been smoothed, or filled up with dust. If now we put some engine oil on the pieces, they will slide very easily. This is because the more minute depressions have been filled up and the whole surface is made comparatively smooth. No matter how carefully we might plane and polish any surface, a microscope would show that it was still a little rough.

One cause of loss of power in the steam engine is friction. In all engines there are so many moving parts that it is of great importance that friction should be reduced as much as possible. This is

done by making the surfaces in contact smooth and of ample size; also making them of different metals and using oils or other lubricants. The effect of the lubricant is to interpose a thin film between the surfaces. This prevents their coming into actual contact. If the oil is too thin or the pressure too great, the lubricant is squeezed out and the metal surfaces come in contact.

Thus we see that there are certain qualities which a lubricant must have. They are as follows :

The lubricant must be sufficiently fluid, so that it will not itself make the bearing run hard.

It must not be too fluid or it will be squeezed out from between the bearing surfaces. If this happens, the bearing will

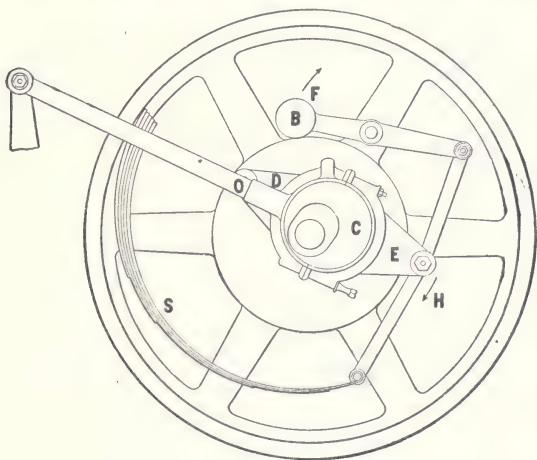


Fig. 43.

immediately begin to heat and cut. The heating will tighten the bearing, and will thus increase the pressure and the cutting.

It must not gum or dry when exposed to the air.

It must not be easily decomposed by the heat generated. If it should be decomposed, it might form substances which would be injurious to the bearings.

It must not take fire easily.

It must contain no acid, and should form no acid in decomposing, as acids corrode the bearings.

Both *mineral* and *animal* oils are used as lubricants. Formerly animal oils were used entirely, but they were likely to decompose at high temperatures and form acids. It is important in using high-pressure steam to have "high-test oils," that is, oils which will not decompose or volatilize at the temperature of the steam. It was the difficulty of getting such oils which made great trouble when superheated steam was first used. Mineral oils will stand these temperatures very readily, and even if they do decompose, they form no acids.

The Liquid Lubricants, whether of animal, vegetable or mineral origin, may be used for ordinary bearings, but for valves and pistons heavy mineral oils only are suitable.

Solid Lubricants. *Graphite* is used as a lubricant. It is well adapted for heavy pressures when mixed with certain oils. It is especially valuable for heavy pressures and low velocities.

Metalline is a solid compound, containing graphite. It is made in the form of solid cylinders, which are fitted to holes drilled into the surface of the bearing. When a bearing is thus fitted no other lubricant is necessary.

Soapstone in the form of powder and mixed with oil or fat is sometimes used as a lubricant. Soap mixed with graphite or soapstone is often used where wood is in contact with wood or iron.

A preparation called *Fiber Graphite* is used for self-lubricating bearings. It is made of finely divided graphite mixed with fibers of wood. It is pressed in molds and afterward fitted to bearings.

For great pressure at slow speed, graphite, lard, tallow and other solid lubricants are suitable. If the pressure is great and the speed high, castor, sperm and heavy mineral oils are used.

For low pressure and high speed, olive, sperm, rape and refined petroleum give satisfaction.

In ordinary machinery, heavy mineral and vegetable oils and lard oil are good. The relative value of various lubricants depends upon the prevailing conditions. Oil that is suitable for one place might not flow freely enough for another.

The quality of oil is of great importance. In many branches

of industry it is imperative that the machinery run as perfectly as possible. On this account and because of the high cost of machinery, only first-class oil should be used. The cylinder oil especially, should be high grade, because the valves, piston and piston rods are the most delicate parts of the engine.

Engines are lubricated by means of oil cups and wipers placed on the bearings wherever required. They are made in many forms, dependent upon the manufacturers. Commonly the oil cup is made with a tube extending up through the oil. A piece of lampwick or worsted leads from the oil in the cup to the tube. Capillary attraction causes the oil to flow continuously and drip down the tube. When not in use, the lampwick should be withdrawn.

The needle oil cup differs from the capillary oil cup in that a small wire or needle extends through the tube and oil; one end rests on the journal to be lubricated. The needle should fit the tube closely, so that when the machinery is at rest no oil will flow. When revolving, the shaft gives the needle a wabbling motion which makes the oil flow. To increase the supply, a smaller needle is used.

The oil cup shown in Fig. 44 is simple and economical. The opening of the valve is regulated by an adjustable stop. The oil may be seen as it flows drop by drop. The cylindrical portion is made of glass, so that the engineer can see how much oil there is in the cup without opening it.

A form of wiper crank pin oiler is shown in Fig. 45. The oil cup is attached to a bracket. The oil drops from the cup into a sheet of wicking or wire cloth and is removed at each revolution of the crank pin by means of the cup which is attached to the end of the connecting rod.

Fig. 46 shows a centrifugal oiling device. The oil flows from the oil cup through the tube to the small hole in the crank

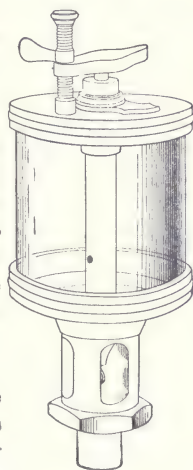


Fig. 44.

pin by centrifugal force. It reaches the bearing surface by means of another small hole.

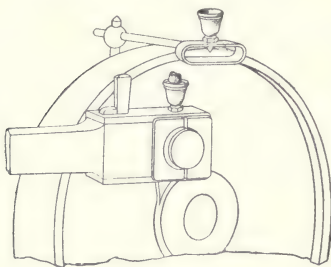


Fig. 45.

In oiling the valve chest and cylinder the lubricant must be introduced against the pressure of the steam. This can be done in several ways, in each of which it is introduced into the steam before it reaches the valve chest and is carried to the surfaces to be lubricated.

The oil can be forced into the steam pipe by a small hand pump or in large engines by an attachment from the engine itself. The supply of oil is, of course, intermittent if the pump is driven by hand, but continuous and economical if driven by the engine.

Sight Feed Lubricators.

The most common device for feeding oil to the cylinder is that which introduces the oil drop by drop into the steam when it is in the steam pipe or steam chest. The oil becomes vaporized and lubricates all the internal surfaces of the engine.

Fig. 47 shows the section of a sight feed lubricator. The reservoir O is filled with oil. The pipe B, which connects with the steam pipe, is partly filled with condensed steam, which flows down the small curved pipe E to the bottom of the chamber O. A small portion of the oil is thus displaced and flows from the top of the reservoir O down the tube F, by the

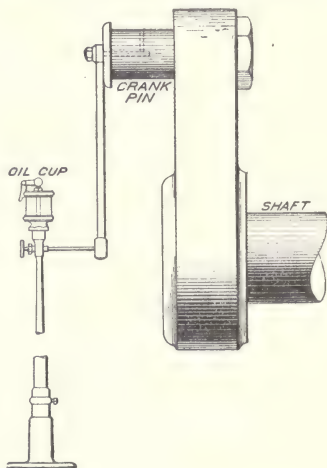


Fig. 46.

regulating valve D, and up through the glass tube S, which is filled with water. It enters the main steam pipe through the connection A. The gage glass G indicates the height of water in the chamber O. To fill the lubricator, close the regulating valve D and the valve in pipe B; the oil chamber can then be drained and filled. If the glass S becomes clogged it may be cleaned by shutting valve D and opening the small valve H. This will allow steam to blow through the glass. After cleaning close valve H and allow glass S to become filled with water before opening the feed valve. The amount of oil fed to the cylinder can be regulated by opening D (Fig. 47) the proper amount. The exact quantity of oil necessary for the engine is not easily determined. For ordinary sizes it is between one drop in two minutes and two drops per minute.

Graphite is an excellent lubricant and can be introduced into the cylinder dry or mixed with some heavy grease. It has been used extensively because of the trouble which cylinder oil gives in the exhaust and in the boilers of condensing engines.

In slow-speed engines it is not hard to attend to the oiling; all the parts are moving slowly and can be readily examined and oiled. Many high-speed engines run so fast that it is impossible to examine the various parts, and special means must be provided for lubricating. It is specially important in high-speed engines that there should be no heating. High-speed engines are generally used for electric lighting, and it is absolutely essential that they be kept running at the required speed to avoid flickering lights. Thus, while there is greater liability to heating in high-

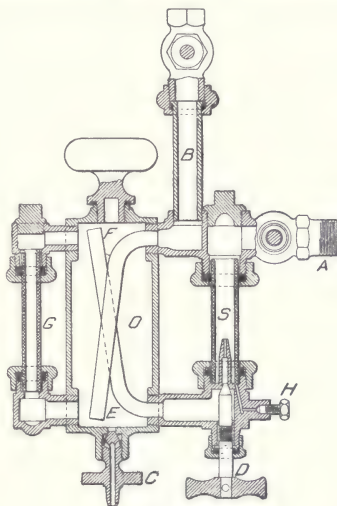


Fig. 47.

speed engines, there is also much greater loss in case heating compels the stopping of an engine.

In order to avoid the danger of forgetting to oil a bearing of a high-speed engine, it is customary to have all the bearings oiled from one place. All the oil is supplied to one reservoir and from this reservoir pipes lead to all bearings. If this is not done, large oil cups are supplied, as a rule, so that oiling need not be attended to as frequently.

In some high-speed engines the moving parts are enclosed and the crank runs in a bath of oil. This secures certain oiling and is very effective. All the bearings may be inside this crank case, so that all are oiled in this way. It is impossible for a careless engineer to overlook one point and so endanger the whole engine.

STEAM TURBINE.

The very earliest records of the steam engine describe a form of steam turbine. It consisted of a hollow sphere, as shown in

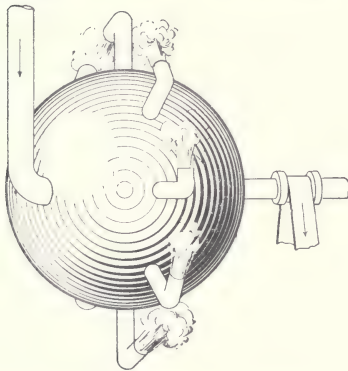


Fig. 48.

Fig. 48, mounted on trunnions, through which steam was admitted to the interior. This steam escaped through pipes bent tangentially to the equator line of the sphere. The force of the escaping steam reacted upon the sphere, causing it to revolve on its trunnions. Many centuries later, in 1629, Branca, an Italian, invented a rotary engine (Fig. 49), in which a jet of steam struck the vanes of a

wheel, and thus forced it around in much the same way that a jet of water acts on a Pelton water-wheel. These engines were of little, if any, practical value, and used an immense quantity of steam.

In 1705 the reciprocating engine was introduced, and by means of Watt's inventions became so efficient that the development of the rotary engine was out of the question. It will be remembered that Watt introduced the expansive use of steam in the reciprocating engine, which at this time could not be accomplished in the rotary engine, and until within the last few years practically nothing was done to develop the turbine.

Since the days of Watt there has been but one important thermodynamic improvement in the reciprocating engine; namely,

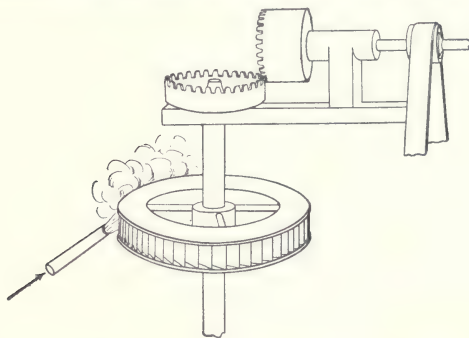


Fig. 49.

the introduction of compound expansion. All other improvements have been in the nature of mechanical devices, and it seems reasonable to suppose that the greatest developments of the future may possibly be in the production of some type other than the reciprocating engine.

In 1883 De Laval invented a successful turbine for running a cream separator, and a short time later Parsons introduced another. Both of these engines employed the expansive force of steam, but each derived this force in a different way.

Since 1883 the development of the turbine has been very rapid. The first engine introduced by De Laval, although far ahead of the earlier forms, was still very wasteful of steam; but now such improvements have been made that their steam consumption compares very favorably with the consumption of good reciprocating engines.

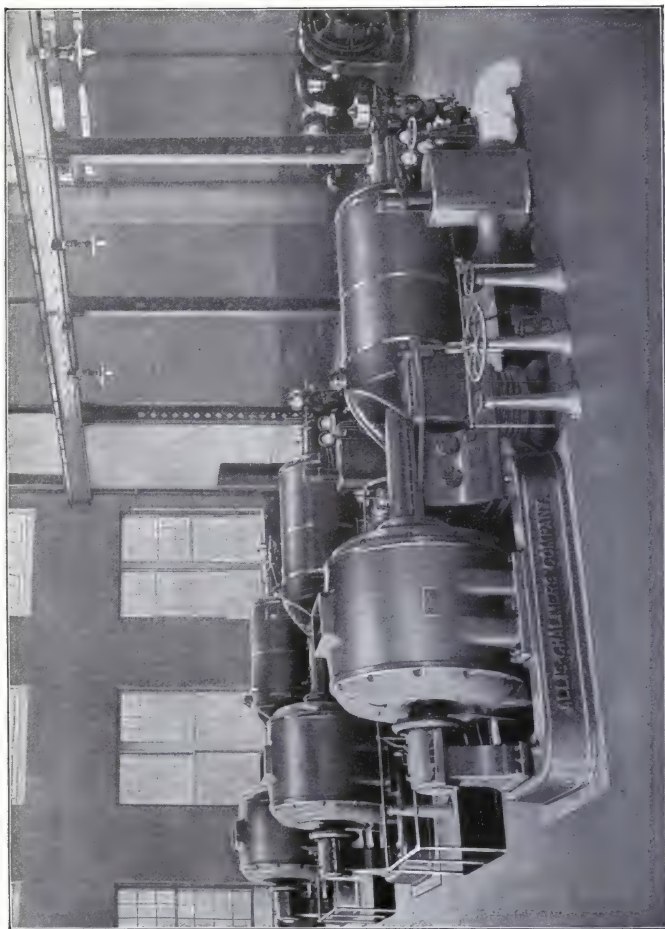
A modern turbine is a tremendously high-speed engine. It does not derive its power from the static force of steam expanding behind a piston, as in the reciprocating engine, but in this case the expanding steam produces kinetic energy of the steam particles. These particles receive a high velocity by virtue of the expansion, and, acting upon the vanes of a wheel, force it around at a high speed of rotation in some such manner as a stream of water rotates a water-wheel.

In the reciprocating engine the expansion produces a force which presses on the piston. In the rotary engine the expansion produces velocity in a jet of steam. This is the fundamental difference between the two forms.

The essential principles of water turbines are equally true of steam turbines. The jet must strike the vanes without a sudden shock, and must leave them in another direction without any sharp deflections. For maximum efficiency the De Laval engine should have a jet velocity equal to twice the linear velocity of a point on the wheel-rim; for the Parsons these velocities should be equal. If the velocity of steam is 8,000 feet per second, it is easily seen that even one-half of this would cause too great a speed of rotation for safety. It would be difficult to build a wheel that would be strong enough to withstand the centrifugal force at this high speed. It becomes necessary, therefore, to reduce the speed to the limits of safety, and run under a slightly less efficiency.

At such high speed the shaft and wheel should be perfectly balanced, in order that its center of gravity may exactly coincide with the axis of rotation. In practice it has been found impossible to balance the shaft perfectly; and in order that it may revolve about its center of gravity, various means are adopted to overcome the rigidity of an ordinary shaft and bearing. This makes high speed of rotation possible without any apparent vibration.

De Laval Turbine. The De Laval turbine shown in Fig. 50 consists of a wheel with suitably shaped buckets, against which a jet of steam is directed. The buckets are on the rim of the wheel and are surrounded by a casing B, which prevents the escape of the steam until it has done its work. A piece of this casing is cut away at A in order to show the buckets. The steam



THREE ALLIS-CHALMERS STEAM TURBINES AND GENERATORS
3,250 Kilowatt Installation. Allis-Chalmers Co., Milwaukee, Wis.

from the nozzle strikes these buckets and is deflected. Thus by the impact of the jet and the reaction due to its deflection, the wheel is caused to revolve at a high speed.

There are usually four nozzles that supply steam to the turbine, one of which is shown in section at D. These nozzles are small at the throat and diverge outward. By making them of the right length and with the proper amount of divergence, the steam can be expanded from the pressure of admission to the

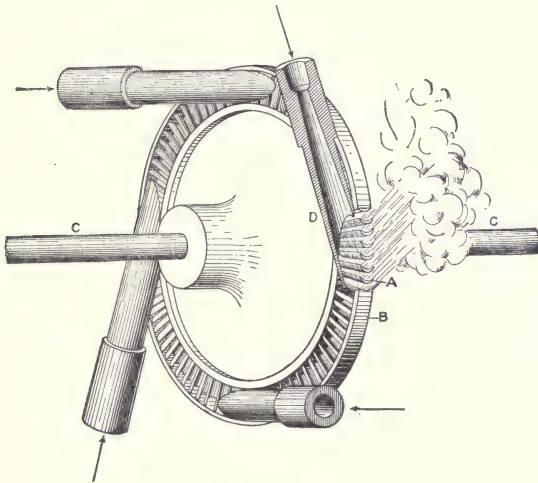


Fig. 50.

pressure of the condenser. Complete expansion is obtained in this diverging nozzle, and the steam leaves it at the exhaust pressure. The steam then works only by virtue of its high velocity.

This turbine has a long, flexible shaft C which can deflect enough to make up for any eccentricity of the center of gravity of the shaft, and thus allow the shaft to revolve about its center of gravity and still have rigid bearings at the end.

Admission is regulated by a throttle-valve, controlled by a fly ball governor.

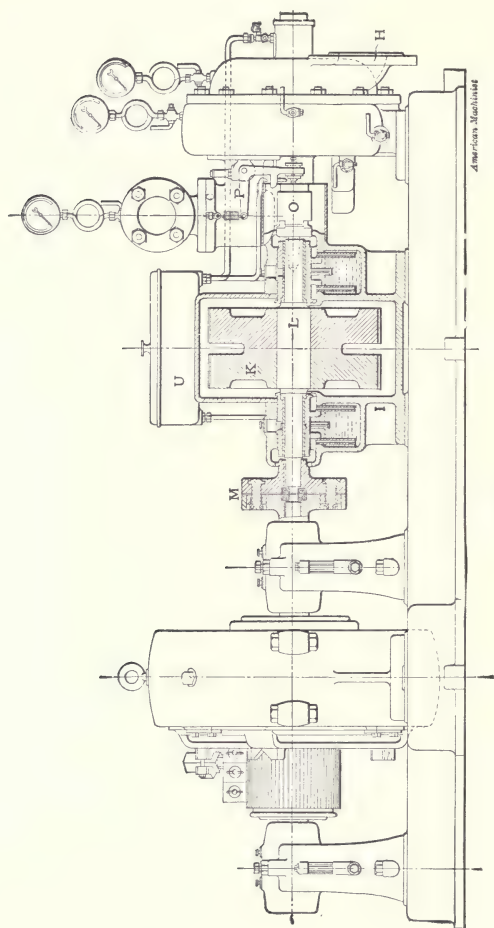


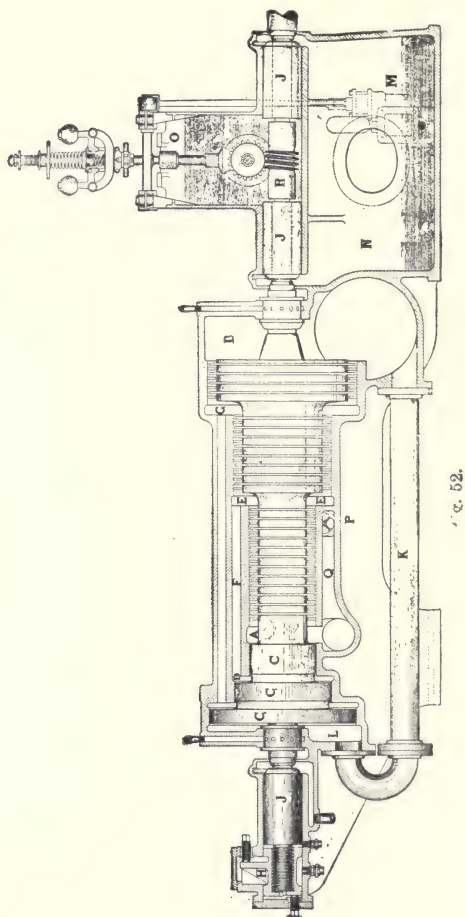
Fig. 51.

Fig. 51 shows a De Laval turbine connected with a generator.

The Parsons Turbine. Fig. 52 is a longitudinal section of a Westinghouse-Parsons turbine. Steam enters the chamber A and passes through the turbine vanes to the exhaust chamber B. The vanes are arranged as shown in Fig. 53, and consist of alternate sets, one stationary, the next movable. The steam strikes one and is deflected to the next; thus the action and the reaction occurring in rapid succession cause the movable sets, which are fixed to the shaft, to rotate at a high speed. As the steam passes the different sets of blades, the volume of the passages increases to correspond with the expansion of the steam. In the De Laval the steam was expanded entirely before reaching the wheel, but in the Parsons the expansion is accomplished in the engine itself. As the steam enters the chamber A (see Fig. 52) it presses on the turbine vanes and it also presses equally and in the opposite direction on C, which is really a piston fixed to the shaft. Thus we see that the pressure to the right is equal to the pressure to the left, and there is no end pressure on the bearing of the shaft. C_1 and C_2 balance the steam pressure in the chambers E and G. At H is a bearing which serves to maintain a correct adjustment of the balance pistons C. There is probably some escape of steam past these balancing pistons, but it is small. The exhaust steam at B presses the turbine toward the left, and would cause an end pressure on the bearing were it not that the pipe K opens a communication between the exhaust chamber B and the back of the balancing pistons, which makes the pressure equal at both ends.

The bearing consists of a gun-metal sleeve surrounded by three concentric tubes. There is a small clearance between these tubes which fills with oil and permits the bearing to run slightly eccentric to counteract any lack of balance in the shaft. Thus the shaft may revolve about its center of gravity, and this oil bearing serves the same purpose as the De Laval flexible shaft.

At P is shown a by-pass valve by means of which live steam may be admitted to the space E, if desirable. Of course this reduces one stage of the expansion, with a corresponding loss of economy, but will increase the power of the turbine. If the condenser fails on a condensing turbine it may still be run at full load by opening the by-pass valve.



Steam is admitted to this turbine in puffs through a reciprocating valve. A fly-ball governor regulates the admission, which is always at boiler pressure.

For electric generators the turbine has many advantages, among them high speed and direct connection. They have small foundations and take up little space; there is slight loss from fric-

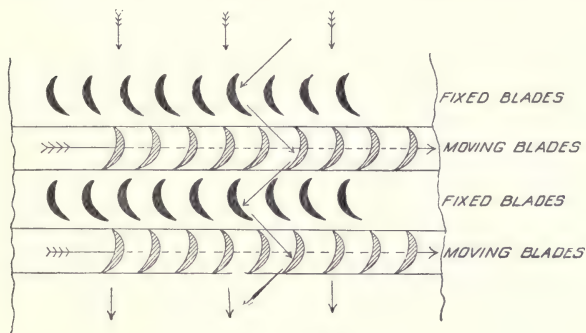
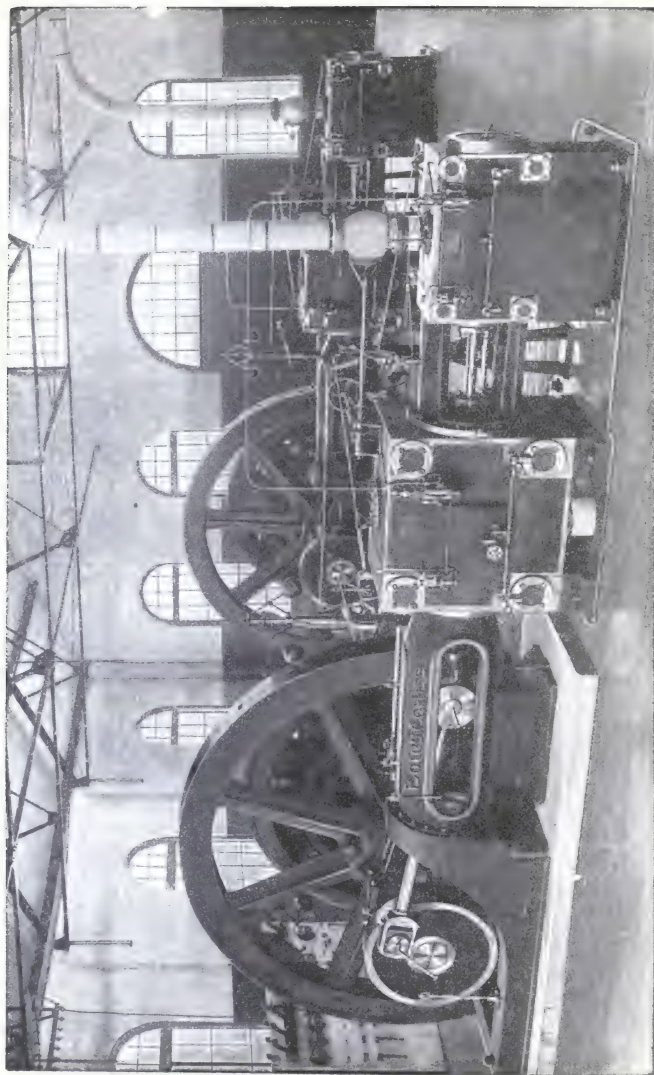


Fig. 53.

tion and few parts. Where slow speed is desired a reciprocating engine is probably the best.



TANDEM COMPOUND BATES CORLISS ENGINES DIRECT-CONNECTED TO 400 K.W. GENERATORS.

Power Station of the Hampton Roads Railway Co., Hampton Va.

THE STEAM ENGINE

PART II

ACTION OF HEAT.

There are several types of heat engines, such as the steam engine, the gas engine, the hot-air engine, etc., each one of which derives its motive power from the heat contained in steam, gas, oil, hot air, or some other substance. Heat is imparted to these substances either by the combustion of fuel in a generator entirely separate from the engine, or by the combustion of a gaseous substance in the cylinder of the engine itself. In the case of the hot-air engine the heat is produced by a fire immediately beneath the engine.

Steam and hot gases have a tendency to expand, because of the heat they contain, thus producing a pressure in all directions. This pressure causes the piston of the engine to move, which allows the gas to expand. As the gas expands, it gives up heat, which is converted into useful work.

We shall now discuss the fundamental principles of the action of heat, and the behavior of gases and vapors, with special reference to the properties of steam and its action in the cylinder of the engine.

If a piece of iron or some other substance is placed in a fire, it becomes hotter than it was before, because heat from the fire has passed into it. If this hot substance is plunged into cold water, or is allowed to remain in a cool place, heat will pass from it, and we say that it becomes cold.

There have been many theories as to what heat is. The accepted theory of to-day is that heat is the result of motion, or we may say it is a form of kinetic energy. Heat is produced, not by the motion of the substance itself (for the hot body may be at rest), but by a rapid vibration of the minute individual particles that make up the body. These minute particles are called molecules. The faster the molecules vibrate, the greater will be their

kinetic energy and the hotter the substance will become. The hottest bodies are those that have the greatest energy of molecular vibration. A hot body can transfer its energy of vibration to another, which in turn becomes hotter than it was before, while the first body loses a part of its energy of vibration and becomes cooler. The terms hot and cold are only comparative terms; one body is hot because it contains a greater degree of heat than another; the other is cold because it contains a less degree of heat than the first. Cold, then, is simply a low degree of heat.

By temperature is meant simply the thermal condition of a body with reference to its capability of transferring heat to other bodies. If two bodies are placed in contact and the first gives more heat to the second than it receives, we say that No. 1 is hotter than No. 2. If the first receives more heat than it gives, No. 2 is hotter than No. 1. If both bodies give and receive the same amount of heat, they are of the same temperature.

According to our theory, it is evident that temperature depends upon the energy of molecular vibration. If the temperature rises, it means that the molecular vibration, and consequently the energy, increases; if the temperature falls, the energy of molecular vibration decreases. Evidently a point must finally be reached when this energy of vibration is zero and the molecules are at rest. At this temperature there is no heat and we call it the absolute zero. It is evident that this zero is very much below the zero of the ordinary scale.

In order to determine just how hot a body is, we must compare its temperature with that of some substance whose degree of heat we know. It would be impossible to keep several bodies at different degrees of heat for comparison, so we must resort to some other means. A simple method is to use some substance whose volume changes a definite amount for a definite change in temperature and always has the same volume for the same temperature. Mercury and alcohol are suitable substances and may be placed in a glass bulb, to which is connected a glass tube of small bore. All the air is drawn out of the tube, and the end is sealed so that the thermometric substance can expand or contract in a vacuum. The tube having been sealed, the bulb is placed in melting ice and the height of the mercury in the tube noted. It is then placed

in steam (or boiling water) at atmospheric pressure and the height of the column again noted. On the Fahrenheit scale the melting point is called 32° , and the boiling point 212° , and the intervening space is divided into 180 equal parts. In the Centigrade scale the melting point is called 0 and the boiling point 100° ; there are 100 equal intervals between them. Thus we see that $180^{\circ} \text{ F} = 100^{\circ} \text{ C}$, or $1^{\circ} \text{ C} = 1.8^{\circ} \text{ F}$.

Example: What is the temperature of 50° C on the F scale?

$$50^{\circ} \text{ C} = 50 \times 1.8 = 90^{\circ} \text{ F above the melting point}$$

$$\text{or } 90 + 32 = 122^{\circ} \text{ F above zero.}$$

In order to compare temperatures, we place the thermometer in contact with the substance whose degree of heat we wish to know and then observe the height of the liquid column in the thermometer. The height of this column depends upon the expansion of the thermometric substance and indicates the intensity of heat, or the temperature as we commonly call it. We use a thermometer to measure the *intensity* of heat, but not the *quantity* of heat.

For measuring the intensity of heat, the degree is the unit; for measuring the quantity of heat we have another unit, which is the amount of heat necessary to raise one pound of water from 61° F to 62° F . This is called the British thermal unit (B. T. U.). To raise one pound of water from 60° F to 62° F , or to raise two pounds from 60° F to 61° F , will require 2 B. T. U.

Suppose we have a small bar of iron heated to a white heat; its temperature will be high, but it will contain a relatively small quantity of heat, that is, it will not require a great many B. T. U. to raise its temperature to this high point. But suppose this same number of heat units were transferred to a ton of iron; the temperature would scarcely be changed, for the infinitely greater number of particles would have a correspondingly less number of vibrations.

Experience teaches us that when a rifle-ball strikes a target it stops, and that the energy which it possesses by virtue of its bodily motion is suddenly transformed into energy of molecular motion. Energy is indestructible; the molecular motion of the impinging body is at once increased and heat is developed. In general, whenever moving bodies are brought to rest, either sud-

denly as by impact or gradually as by friction, the kinetic energy of the moving mass is transformed into molecular kinetic energy, and we say that the bodies become heated.

We have now seen that we can transform energy of motion into heat. Also by means of suitable apparatus we can transform heat into energy of motion. Heat is the lowest form of energy, and while it is comparatively easy to transform other forms into heat, it is not as easy to change heat into the higher forms of energy. The principle of the transformation is simple enough, but if we are to have an efficient engine, we must be able to extract practically all of the heat from the working substance. This, however, is impossible, because the ordinary ranges of temperature used in practice are so far removed from the absolute zero of temperature, that with the most perfect machines we can at best recover but a fraction of the heat; the rest passing out of the engine.

The transformation is accomplished by means of a working substance which passes from the temperature of a heat generator into a refrigerator; the heat given up during the change is transformed into work. By the term refrigerator we mean the low temperature of the working substance at exhaust. The greater the temperature of the source of heat, or the lower the temperature of the refrigerator, the greater will be the amount of heat that can be abstracted and converted into useful work. If the temperature of the refrigerator could be reduced to the absolute zero, all the heat would be removed from the working substance, and the only loss would be that due to mechanical imperfections of the engine; but since the absolute zero is 461° below the zero on the Fahrenheit scale, or 493° below the freezing point, we must at best allow a relatively large amount of heat to pass out of the engine into the refrigerator without having done any work at all.

The unit of work is the foot-pound; that is, the work done in raising one pound one foot. By means of careful experiments it has been determined that for every 778 foot-pounds of work transformed into heat there is developed one B. T. U. This value 778 is known as the Mechanical Equivalent of heat. It means that one B. T. U. and 778 foot-pounds of work are mutually interchangeable.

EXPANSION OF GASES.

A perfect gas, strictly speaking, is one that cannot be liquefied ; ordinarily, however, we apply the term to those gases that can be liquefied only with great difficulty, that is, under extreme pressure and a great reduction of temperature. For every perfect gas there is a definite relation between the pressure and volume, and what is known as Boyle's Law has been found to hold true, *viz.* : The pressure of a perfect gas at constant temperature varies *inversely* as the volume ; that is, if the temperature remains constant the pressure becomes less as fast as the volume becomes greater, or conversely the volume becomes less as fast as the pressure becomes greater. Now if the pressure becomes twice as great, the volume becomes half as large, and if the volume becomes three times as great, the pressure will be only one-third as much. Hence we see that however the pressure may vary, the volume will change in such a way that the pressure multiplied by the volume will always be constant, provided the temperature remains the same.

This simple law is expressed thus :

$$P \times V = C$$

In which

P = pressure in pounds per square inch (absolute)

V = volume in cubic feet

C = a constant which has different values for different gases.

Gases that are not easily liquefied, such as hydrogen, oxygen and air, follow this law fairly well, but those that are easily liquefied, such as steam and ammonia, do not follow it at all.

The value of C is not the same for all gases, and as no gas is, strictly speaking, a perfect gas, its value varies slightly with the temperature. For air, which is nearly a perfect gas, its value is 182.08 at 32° F.

Example.—What is the absolute pressure per square inch of one pound of air if the temperature is 32° F, and the volume 4.129 cubic feet?

$$V = 4.129$$

$$P \times V = C \quad \text{or} \quad P = \frac{C}{V} = \frac{182.08}{4.129}$$

$$P = 44.1 \text{ pounds per square inch, absolute.}$$

Let us discuss this law by means of a diagram. Fig. 1 is drawn with OY and OX at right angles to each other; pressures are measured to any convenient scale on OY and volumes on OX . OC represents 12.387 cubic feet and OD represents 14.7 pounds, since $P \times V = 182.08$. OF represents 24.774 cubic feet and OG 7.35 pounds. In like manner OL represents 6.19 cubic feet and ON 29.41 pounds. Then if we draw perpendiculars to OX at L , C and F , and perpendiculars to OY at N , D and G , they will meet in the points M , E and H . The curve AB is drawn through

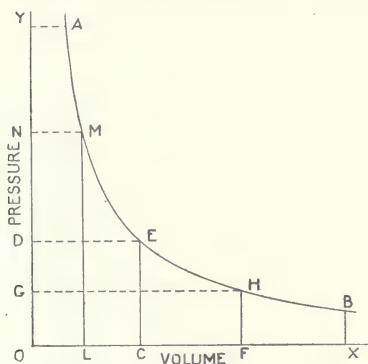


Fig. 1.

these points. Then for any pressure we can find the corresponding volume or *vice versa*. The area of the rectangle $ODEC$ equals that of the rectangle $OGHF$ and also that of the rectangle $ONML$. That this is so is readily seen from the fact that the product of the pressure and its corresponding volume is constant.

If we plot a curve using this equation we will get a rectangular hyperbola, as shown in "Steam Engine Indicators." This curve is called an isothermal curve or curve of equal temperatures.

If the volume of a perfect gas remains constant, the pressure will vary as the temperature. Or, if the pressure remains constant, the volume will vary as the temperature. This is known as the Law of Charles. From this we see that, as the temperature decreases, either the pressure or the volume will decrease a proportionate amount, and this must continue as long as there is any heat in the gas; finally a low temperature (the absolute zero) will be reached, where there is no more heat, and consequently either the pressure or the volume must be zero, provided this law holds true at such a very low temperature.

The law states that any change of pressure or volume is pro-

portional to the change in temperature, that is, the new volume is equal to the first volume plus or minus some fractional part of this volume, called the coefficient of expansion, multiplied by the change in temperature. We may express the formula thus:

$$\begin{aligned} P_1 V_1 &= P V + (P V) k \times t \\ &= P V (1 + k t). \end{aligned}$$

Where P_1 and V_1 are the new pressure and volume

P and V are the first pressure and volume

k is the coefficient of expansion for the gas

t is the change in temperature.

From Boyle's Law we know that $P V = C$; hence we may write the above equation:

$$\begin{aligned} P_1 V_1 &= C (1 + k t) \\ &= k C \left(\frac{1}{k} + t \right) \end{aligned}$$

From careful experiments on the expansion of air, Regnault determined the value of the coefficient k to be .003654 for Centigrade units. This value is constant between freezing point, 0, and boiling point, 100°. Substituting this value of k in our last equation, we have:

$$\begin{aligned} P_1 V_1 &= k C \left(\frac{1}{.003654} + t \right) \\ &= k C (273.7 + t). \end{aligned}$$

Now if we should make the change of temperature, t , equal to -273.7° we should have:

$$\begin{aligned} P_1 V_1 &= k C (273.7 - 273.7) = 0 \\ P_1 V_1 &= 0. \end{aligned}$$

Therefore we must have reached the absolute zero at -273.7° below the freezing point, because here $P_1 \times V_1$ has reached 0, as we have previously seen must be the case at the absolute zero. $273.7 \times 1.8 = 492.7^\circ$ for the F scale. Therefore the absolute zero is $492.7^\circ - 32^\circ = 460.7^\circ$ below zero on the Fahrenheit scale. For ordinary work 461 will be sufficiently accurate.

If we let P = absolute pressure in pounds per square inch

V = volume of one pound in cubic feet

T = absolute temperature on Fahrenheit scale,

Then,

$$P V = C T \text{ } \mathcal{L}.$$

$k C$ equals .3693 for air.

Example.—If one pound of air occupies 16.606 cubic feet at a pressure of 14.7 pounds per square inch, what is its temperature?

$$\begin{aligned}
 P \times V &= C \times T \\
 14.7 \times 16.606 &= .3693 \times T \\
 T &= \frac{244.108}{.3693} \\
 T &= 661.
 \end{aligned}$$

This value 661 is absolute temperature, and to find the Fahrenheit temperature 461 must be subtracted from it. Thus, $661^{\circ} - 461^{\circ} = 200^{\circ} \text{ F.}$

Saturated Vapor. The process of converting a liquid into a vapor is known as vaporization; the product thus formed is readily condensed and therefore does not follow the laws of perfect gases at all. A dry saturated vapor is one that has just enough heat in it to keep it in the form of a vapor; if we add more heat it becomes superheated. A superheated vapor may lose

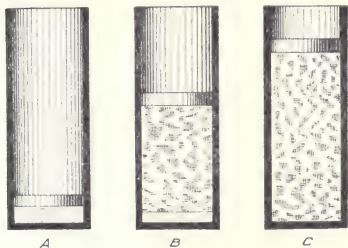


Fig. 2.

a part of its heat without condensation; a saturated vapor cannot. When a saturated vapor loses a part of its heat some of it will condense and we say that the vapor is wet.

Steam is simply the vapor from water and we shall confine our discussion to this alone. Suppose we have a vertical cylinder, as shown in Fig. 2, fitted with a light piston free to move up and down, yet so constructed that it may be loaded at will. Suppose that there is one pound of water at a temperature of 32° F in the bottom of this cylinder, and that the piston rests upon its surface. Now, if we apply heat by means of a gas flame or fire, we shall notice the following effects:

First.—The temperature of the water will gradually rise until it reaches the temperature at which steam is formed. This temperature will depend upon the pressure, or the load on the

piston. If the piston is very light, we may neglect its weight and consider that there is simply the atmospheric pressure of 14.7 pounds per square inch acting on the water surface. At this pressure steam will begin to form at 212° F.

Second.—As soon as 212° F is reached, steam will begin to form and the piston will steadily rise, but no matter how hot the fire may be, the temperature of both water and steam will remain at 212° until all the water is evaporated. We had one pound of water at 32° F and at 14.7 pounds absolute pressure, and found that steam formed at a temperature of 212° F and remained at that temperature. We added 180.9 B. T. U., the heat of the liquid, to bring the water from 32° to the boiling point. To convert water at 212° into steam at 212°, we added 965.7 B. T. U. more. This quantity, known as the latent heat, or heat of vaporization, makes the total heat 1,146.6 B. T. U. If we should measure the volume carefully after all the water was evaporated, we should find that there was just 26.36 cubic feet of dry saturated steam. We had one pound of water, and therefore must have one pound of steam, for none of it could escape; hence one cubic foot will weigh $\frac{1}{26.36} = 0.03794$ pounds, which is known as the density of steam at 14.7 pounds absolute pressure or 212° F. In the Table of Properties of Saturated Steam (see page 14) all these quantities are found in the order given and at the pressure of 14.7 pounds above vacuum.

Suppose now we place a weight of 85.3 pounds on the piston. The pressure is 85.3 pounds plus 14.7 pounds, or 100 pounds absolute. We shall now find that no steam will form until a temperature of 327.58° is reached. Starting with water at 32°, it will be necessary to add 297.9 B. T. U. before a temperature of 327.58° is reached, and also we must add 884.0 B. T. U. more to vaporize it, making a total heat of 1,181.9 B. T. U. Under this greater pressure the steam occupies a volume of only 4.403 cubic feet, or one cubic foot of it weighs $\frac{1}{4.403} = 0.2271$ pound.

Of course it would be impossible to determine all these different quantities by actual experiment, and at all pressures varying from vacuum to the high pressures, used in water-tube boilers.

Fortunately we are able to compute them all from equations which have been carefully determined by experiment. If saturated steam were a perfect gas, we could easily calculate all the relations of pressure, volume and temperature from the equation $PV = CT$, but steam is so far removed from the state of a perfect gas that these relations do not hold, and the true equations become very complex. The following equation proposed by Rankine is one of the simplest, and gives fairly good results:

$$\log_{10} P = A - \frac{B}{T} - \frac{C}{T^2}$$

in which P = pressure in pounds per square inch above vacuum.

$A = 6.1007$

$B = 2,732$

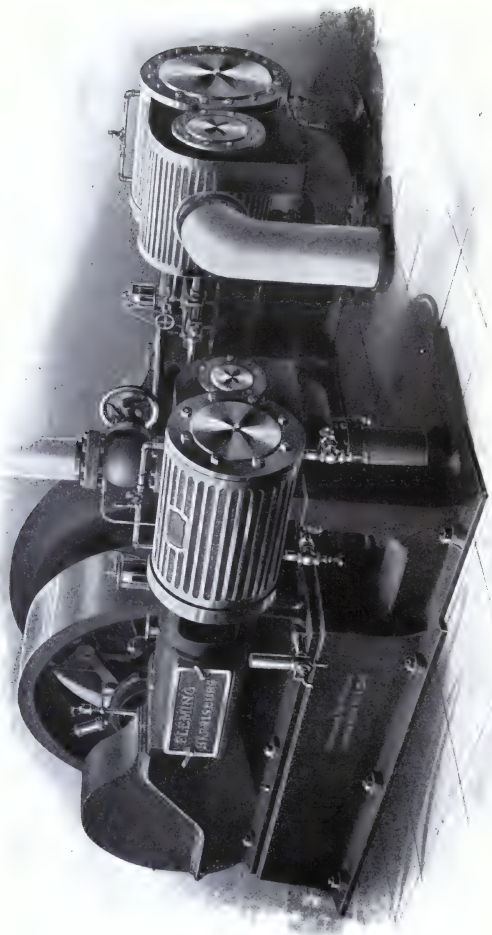
$C = 396,945$

T = absolute temperature in Fahrenheit degrees.

By the aid of such equations as this, all the different quantities found in the steam tables may be calculated. These equations are based on careful experiments and give very satisfactory results.

Steam Tables. We have already seen that any change in the temperature of saturated steam produces a change of pressure, and that every change of pressure corresponds to a certain change in temperature. There are several properties of saturated steam that depend upon the temperature and pressure; and the values of all these different properties when arranged for all temperatures and pressures are called Steam Tables. The following are the principal items that are found in the tables:

1. The absolute pressure in pounds per square inch; it is equal to the gage pressure plus the atmospheric pressure of 14.7 pounds.
2. The temperature of the steam, or boiling water, at the corresponding pressure.
3. The heat of the liquid; or the number of B. T. U. necessary to raise one pound of water from 32° F to the boiling point corresponding to the given pressure.
4. The heat of vaporization, or the latent heat; this is the number of B. T. U. necessary to change one pound of water, at the boiling point, into dry saturated steam at the same temperature and pressure.



FLEMING-HARRISBURG SIDE-CRANK CROSS-COMPOUND ENGINE
General Construction. Direct-Connected Style. Harrisburg Foundry & Machine Co., Harrisburg, Penna.

5. The total heat or the number of B. T. U. necessary to change one pound of water from 32° F into steam at the given temperature or pressure. The total heat is evidently equal to the sum of the heat of the liquid and the heat of vaporization.

6. The density of the steam; that is, the weight in pounds of one cubic foot of steam at the given temperature or pressure.

7. The specific volume; or volume in cubic feet of one pound of steam at the required temperature or pressure. Evidently the specific volume is equal to $\frac{1}{\text{density}}$.

All these properties have been calculated by means of various formulas which have been deduced from the results of actual experiment. There are several formulas for the temperatures and pressures of steam; as some computers have used one and some another, there is likely to be a slight discrepancy between the tables computed by different authors. Rankin's formula, already given, is the simplest, but is not generally considered to be quite as accurate as some of the later ones; it has probably been used, however, more than any other.

The total heat may be calculated by means of the formula

$$H = 1,091.7 + 0.305 (t - 32) \quad [1]$$

in which H = total heat

t = temperature in degrees Fahrenheit.

The heat of the liquid is equal to q .

$$q = t + 0.00002 t^2 + 0.0000003 t^3 \quad [2]$$

in which t = temperature in degrees Centigrade.

These constants are for use in the Centigrade system only. To calculate the heat of the liquid for any Fahrenheit temperature it is necessary to change the Fahrenheit into equivalent Centigrade degrees and then substitute in the above formula.

The formula used for the calculation of specific volume is too complex for consideration here, but the relation of pressure and volume may be approximately expressed by means of an equation of the form $P V^n = C$, in which n is an exponent, C a constant, P the absolute pressure, and V the specific volume; n is usually taken to be $\frac{17}{16}$, and C to be 475.

On pages 14 and 15 are given tables of the properties of

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure in pounds persq.in. above vacuum.	Temperature in degrees Fahrenheit.	Heat in liquid from 32° in units.	Heat of vaporization, or latent heat in heat units	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of 1 pound in cubic feet.	Total pressure above vacuum.
1	101.99	70.0	1043.0	1113.1	0.00299	334.5	1
2	126.27	94.4	1026.1	1120.5	0.00576	173.8	2
3	141.62	109.8	1015.3	1125.1	0.00844	118.5	3
4	153.09	121.4	1007.2	1128.6	0.01107	90.31	4
5	162.34	130.7	1000.8	1131.5	0.01366	73.21	5
6	170.14	138.6	995.2	1133.8	0.01622	61.67	6
7	176.90	145.4	990.5	1135.9	0.01874	53.37	7
8	182.92	151.5	986.2	1137.7	0.02125	47.06	8
9	188.33	156.9	982.5	1139.4	0.02374	42.12	9
10	193.25	161.9	979.0	1140.9	0.02621	38.15	10
14 7	212.00	180.9	965.7	1146.6	0.03794	26.36	14.7
15	213.03	181.8	965.1	1146.9	0.03826	26.14	15
20	227.95	196.9	954.6	1151.5	0.05023	19.91	20
25	240.04	209.1	946.0	1155.1	0.06199	16.13	25
30	250.27	219.4	938.9	1158.3	0.07360	13.59	30
35	259.19	228.4	932.6	1161.0	0.08508	11.75	35
40	267.13	236.4	927.0	1163.4	0.09644	10.37	40
45	274.29	243.6	922.0	1165.6	0.1077	9.287	45
50	280.85	250.2	917.4	1167.6	0.1188	8.414	50
55	286.89	256.3	913.1	1169.4	0.1299	7.696	55
60	292.51	261.9	909.3	1171.2	0.1409	7.097	60
65	297.77	267.2	905.5	1172.7	0.1519	6.583	65
70	302.71	272.2	902.1	1174.3	0.1628	6.143	70
75	307.38	276.9	898.8	1175.7	0.1736	5.762	75
80	311.80	281.4	895.6	1177.0	0.1843	5.426	80
85	316.02	285.8	892.5	1178.3	0.1951	5.126	85
90	320.04	290.0	889.6	1179.6	0.2058	4.859	90
95	323.89	294.0	886.7	1180.7	0.2165	4.619	95
100	327.58	297.9	884.0	1181.9	0.2271	4.403	100
105	331.13	301.6	881.3	1182.9	0.2378	4.205	105
110	334.56	305.2	878.8	1184.0	0.2484	4.026	110
115	337.86	308.7	876.3	1185.0	0.2589	3.862	115
120	341.05	312.0	874.0	1186.0	0.2695	3.711	120
125	344.13	315.2	871.7	1186.9	0.2800	3.571	125
130	347.12	318.4	869.4	1187.8	0.2904	3.444	130
140	352.85	324.4	865.1	1189.5	0.3113	3.212	140
150	358.26	330.0	861.2	1191.2	0.3321	3.011	150
160	363.40	335.4	857.4	1192.8	0.3530	2.833	160
170	368.29	340.5	853.8	1194.3	0.3737	2.676	170
180	372.97	345.4	850.3	1195.7	0.3945	2.535	180
190	377.44	350.1	847.0	1197.1	0.4153	2.408	190
200	381.73	354.6	843.8	1198.4	0.4359	2.294	200
225	391.79	365.1	836.3	1201.4	0.4876	2.051	225
250	400.99	374.7	829.5	1204.2	0.5393	1.854	250
275	409.50	383.6	823.2	1206.8	0.5913	1.691	275
300	417.42	391.9	817.4	1209.3	0.644	1.553	300
325	424.82	399.6	811.9	1211.5	0.696	1.437	325
350	431.90	406.9	806.8	1213.7	0.748	1.337	350
375	438.40	414.2	801.5	1215.7	0.800	1.250	375
400	445.15	421.4	796.3	1217.7	0.853	1.172	400
500	466.57	444.3	779.9	1224.2	1.065	.939	500

TABLE OF PROPERTIES OF SATURATED STEAM.

Temperature in degrees Fahrenheit.	Total pressure above vacuum.	Heat in liquid from 32° in units.	Heat of vaporization, or latent heat in heat units.	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of one pound in cubic feet.	Temperature in degrees Fahrenheit.
32	0.089	0.	1091.7	1091.7	0.0003	3387.	32
60	0.254	28.12	1072.1	1100.2	0.0008	1234.	60
90	0.692	58.04	1051.4	1109.4	0.0021	474.6	90
120	1.683	88.10	1034.4	1118.5	0.0049	204.4	120
140	2.877	108.2	1016.4	1124.6	0.0081	123.2	140
150	3.706	118.3	1009.4	1127.7	0.0103	97.03	150
160	4.729	128.4	1002.3	1130.7	0.0130	77.14	160
170	5.98	138.5	995.3	1133.8	0.0162	61.85	170
180	7.50	148.5	988.3	1136.8	0.0200	50.01	180
190	9.33	158.6	981.3	1139.9	0.0245	40.73	190
200	11.52	168.7	974.2	1142.9	0.0299	33.40	200
210	14.12	178.8	967.2	1146.0	0.0363	27.57	210
220	17.19	188.9	960.1	1149.0	0.0435	22.98	220
225	18.91	193.9	956.7	1150.6	0.0476	20.99	225
230	20.78	198.9	953.2	1152.1	0.0521	19.20	230
235	22.80	204.0	949.6	1153.6	0.0569	17.59	235
240	24.98	209.0	946.1	1155.1	0.0619	16.14	240
245	27.33	214.1	942.6	1156.7	0.0674	14.83	245
250	29.86	219.1	939.1	1158.2	0.0733	13.65	250
255	32.57	224.1	935.6	1159.7	0.0795	12.57	255
260	35.48	229.2	932.0	1161.2	0.0862	11.60	260
265	38.60	234.2	928.6	1162.8	0.0933	10.72	265
270	41.94	239.3	925.0	1164.3	0.1008	9.918	270
275	45.51	244.3	921.5	1165.8	0.1088	9.187	275
280	49.33	249.3	918.0	1167.3	0.1173	8.521	280
285	53.39	254.4	914.5	1168.9	0.1264	7.913	285
290	57.72	259.4	911.0	1170.4	0.1359	7.356	290
295	62.33	264.4	907.4	1171.9	0.1461	6.847	295
300	67.22	269.5	903.9	1173.4	0.1567	6.380	300
305	72.42	274.5	900.5	1175.0	0.1680	5.952	305
310	77.83	279.6	896.9	1176.5	0.1799	5.558	310
315	83.77	284.8	893.2	1178.0	0.1925	5.195	315
320	89.95	290.0	889.5	1179.5	0.2058	4.861	320
325	96.48	295.2	885.9	1181.1	0.2197	4.552	325
330	103.38	300.5	882.1	1182.6	0.2343	4.267	330
335	110.66	305.7	878.4	1184.1	0.2498	4.004	335
340	118.34	310.9	874.7	1185.6	0.2660	3.760	340
345	126.43	316.1	871.1	1187.2	0.2830	3.534	345
350	134.95	321.4	867.3	1188.7	0.3008	3.324	350
355	143.91	326.6	863.6	1190.2	0.3195	3.130	355
360	153.33	331.8	859.9	1191.7	0.3391	2.949	360
365	163.22	337.1	856.2	1193.3	0.3597	2.780	365
370	173.60	342.3	852.5	1194.8	0.3812	2.623	370
375	184.49	347.5	848.8	1196.3	0.4038	2.476	375
380	195.91	352.8	845.0	1197.8	0.4276	2.338	380
385	207.87	358.0	841.4	1199.4	0.4521	2.212	385
390	220.39	363.2	837.7	1200.9	0.4780	2.092	390
395	233.50	368.4	834.0	1202.4	0.5051	1.980	395
400	247.21	373.7	830.2	1203.9	0.5336	1.874	400
405	261.55	378.9	826.6	1205.5	0.5633	1.775	405
410	276.54	384.1	822.9	1207.0	0.5945	1.682	410
415	292.21	389.4	819.1	1208.5	0.6270	1.595	415
420	308.57	394.6	815.4	1210.0	0.6610	1.512	420
425	325.65	399.8	811.8	1211.6	0.6970	1.434	425

saturated steam, one varying with the temperatures, the other with the pressures. The tables are made out in five unit intervals; intermediate points are proportional.

Example.—Suppose we wish to find the total heat corresponding to a pressure of 112.3 pounds (gauge). We first add 14.7 to the 112.3 and get 127 pounds absolute pressure. The total heat of 1 pound of steam at 125 pounds pressure is 1,186.9. The total heat at 130 pounds pressure is 1,187.8.

$$\text{Difference for 5 pounds} = 1,187.8 - 1,186.9 = .9$$

$$\text{Difference for 1 pound} = .9 \div 5 = .18$$

$$\text{Difference for 2 pounds} = 2 \times .18 = .36$$

The total heat for 127 pounds is :

$$\text{Total heat at 125 pounds} = 1,186.9$$

$$\text{Difference for 2 pounds} = \underline{\quad .36 \quad}$$

$$\text{Total heat at 127 pounds} = 1,187.26$$

This method is called interpolation, and in many complete tables the differences for the intervals are given to facilitate the work.

If steam tables are not at hand, there are several approximate formulas that may be used for rough calculations and estimates, but it must be borne in mind that results obtained by the use of these equations are not strictly accurate, and should not be used if the regular tables can be had.

Probably the relation of temperature and pressure will be most frequently needed. If the gage pressure is between 20 pounds and 100 pounds

$$t = 14 \sqrt{p} + 198 \text{ approximately} \quad [3]$$

where t = temperature in degrees Fahrenheit
 p = gage pressure in pounds per square inch.

For pressures over 100 pounds per square inch (gage) we must modify the equation thus :

$$t = 14 \sqrt{p} + 198 - \left(\frac{p - 100}{11} \right) \quad [4]$$

These equations will cover a range of pressures from 20 to 340 pounds, and give an error of less than $1\frac{1}{2}^{\circ}$ in nearly all cases. From 35 pounds per square inch to 100 pounds the error is generally less than one-half of one degree.

For pressures below 20 pounds use the constant 196 instead of 198.

The latent heat may be approximately expressed by the formula,

$$l = 1,114 - .7 t \quad [5]$$

in which

l = latent heat

t = temperature in degrees F.

This formula gives very good results for temperatures less than 320° , corresponding to a gage pressure of about 75 pounds. Above this pressure the formula gives slightly larger results than are found in the steam tables. At 250 pounds (gage) the formula gives 829.8 and the steam tables 825.8, so that the error will not be large in any case.

We have defined a B. T. U. as the amount of heat necessary to raise one pound of water from 61° F to 62° F. The specific heat of water is nearly constant over ordinary ranges of temperature and at 400° we find the heat of the liquid from the tables to be 373.7 B. T. U. By definition, the heat of the liquid at 32° is zero, so that a rise of 368° in temperature requires 373.7 B. T. U. If we consider the heat of the liquid proportional to the rise in temperature, our error will be 5.7 units of heat in 400° . At lower temperatures the error is much smaller, so that we may express the heat of the liquid approximately by the formula,

$$h = t - 32 \quad [6]$$

in which

h = the heat of the liquid

t = temperature in degrees as before.

The total heat is equal to the sum of the last two, or

$$H = h + l \quad [7]$$

The relation of pressure and volume of steam may be approximately expressed by the equation

$$P V^n = C, \text{ or } V^n = \frac{C}{P}$$

In which P = absolute pressure

V = specific volume

n = an exponent = $\frac{17}{16}$

C = a constant = 475 nearly.

We may then write the equation

$$V^{\frac{17}{16}} = \frac{475}{P} \text{ or } V = \sqrt[17]{\frac{475}{P}} \quad [8]$$

This is called the equation of constant steam weight; it may be solved by the aid of logarithms.

The density can of course be determined from the specific volume.

Let us apply these approximate formulas to a specific case and see how the results compare with the actual quantities given in the steam tables. For this purpose we will suppose steam at 70.3 pounds gage pressure or 85 pounds absolute.

$$\begin{aligned} \text{Equation (3)} \quad t &= 14 \sqrt{p} + 198 \\ &= 14 \sqrt{70.3} + 198 = 315.40^\circ \end{aligned}$$

$$\text{From steam tables, temperature} = 316.02^\circ$$

$$\begin{aligned} \text{Equation (5)} \quad l &= 1,114 - .7 t \\ &= 1,114 - (.7 \times 315.4) = 893.2 \end{aligned}$$

$$\text{From steam tables, latent heat} = 892.5$$

$$\begin{aligned} \text{Equation (6)} \quad h &= t - 32^\circ \\ &= 315.6 - 32 = 283.4 \end{aligned}$$

$$\text{From tables, heat in the liquid} = 285.8$$

$$\begin{aligned} \text{Equation (7)} \quad H &= h + l \\ &= 285.8 + 893.2 = 1,179.0 \end{aligned}$$

$$\text{From tables, total heat} = 1,178.3$$

$$\begin{aligned} \text{Equation (8)} \quad V &= \sqrt[17]{\frac{475}{P}} \\ V &= \sqrt[17]{\frac{475}{85}} = 5.100 \end{aligned}$$

$$\text{From tables, specific volume} = 5.125.$$

A comparison of these results shows that these formulas cannot be used when accuracy is sought; but if only approximate results are desired they will be found satisfactory. Whenever possible the steam tables should be used in preference to any approximations.

Superheated Vapor. We have seen that a saturated vapor contains just enough heat to keep it in the form of a vapor; if it

loses heat it will condense. A superheated vapor is one that has been heated after vaporization; it can lose this extra heat before any condensation will take place. A vapor in contact with its liquid is saturated; one heated after removal from the liquid is superheated.

For saturated steam there is a fixed temperature for every pressure. If we know either the pressure or the temperature, we can find the other in the steam tables. For instance, if the gage pressure of a boiler is 60.3 pounds and we wish to know the temperature, we simply add atmospheric pressure and turn to our tables and find it to be 307° (about).

With superheated steam the case is entirely different, for there is no longer the same direct relation between the temperature and pressure. In fact, the relation between temperature and pressure of superheated steam depends upon the amount of superheating. Superheated steam at 60.3 pounds gage pressure may have a temperature considerably above 307° F. At a given pressure the temperature and volume of a given weight of superheated steam are always greater than the temperature and volume of the same weight of saturated steam. The properties of superheated steam at given pressure are not constant as is the case with saturated steam.

If superheated steam were a perfect gas, we could determine the relation of P , V and T by the equation $PV = CT$; but superheated steam is not a perfect gas, hence we must modify our equation. By experiment it has been determined that the following equation is nearly correct:

$$PV = 93.5 T - 971 P^{\frac{1}{4}}$$

In which P = absolute pressure in pounds per square foot

T = absolute temperature

V = volume of 1 pound in cubic feet.

THE STEAM ENGINE.

We have studied the action and formation of steam, and now we shall consider its application to the steam engine. We know that steam contains a great deal of heat, and that heat can be converted into work by allowing a working substance to pass from the high temperature of the heat generator to the lower tempera-

ture of the refrigerator, during this change giving up heat, which is transformed into work. There are several forms of heat engines, all of which convert the heat contained in some substance into work. At the present time the steam engine is the most important. When of good size and properly designed and run, it is as economical as any other heat engine, and it can be more easily controlled and regulated. We shall consider first the theoretically perfect engine and then the modifications that go to make up the steam engine of to-day.

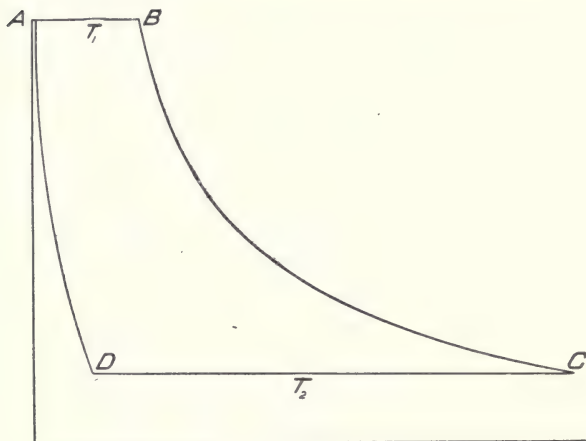


Fig. 3.

The theoretical engine (Fig. 3) is supposed to receive heat from the generator at constant temperature T_1 until communication is interrupted at B. The working substance expands to C without losing or gaining any heat from external sources until the temperature of the refrigerator is reached. The engine now rejects heat at the constant temperature T_2 of the refrigerator and then compresses the working substance without loss or gain in the quantity of heat until the temperature of the heat generator is reached. These are ideal conditions, and, if fulfilled, the efficiency of the perfect engine will depend only on the difference between the tem-

perature at which heat is received and rejected, or, in other words, it depends only upon the difference in temperature between the generator and the refrigerator.

If T_1 = absolute temperature of heat received and

T_2 = absolute temperature of heat rejected, then the thermal efficiency, E , of the engine will be represented by the formula,

$$E = \frac{T_1 - T_2}{T_1}$$

Or, in other words, the efficiency equals the absolute temperature of the heat rejected, subtracted from the absolute temperature of the heat received, and the remainder divided by the absolute temperature of the heat received.

Suppose an engine is supplied with steam at 120 pounds absolute pressure, and the exhaust is at atmospheric pressure. What is the thermal efficiency?

The absolute temperature corresponding to 120 pounds pressure is $341.05 + 461 = 802.05^\circ$, and the absolute temperature of the exhaust is $212 + 461 = 673^\circ$.

$$\text{Then } E = \frac{802.05 - 673}{802.05} = .16, \text{ or } 16 \text{ per cent.}$$

In actual engines this efficiency cannot be realized, because the difference between the heat received and the heat rejected is not all converted into useful work. Part of it is lost by radiation, conduction, condensation, leakage and imperfect action of the valves. The cylinder walls of the theoretical engine are supposed to be made of a nonconducting material, while in the actual engine the walls are of metal, which admits of a ready interchange of heat between cylinder and steam. This action of the walls cannot be overcome, and is so important that a failure to consider its influence will lead to serious errors in computations, and no design can be made intelligently if based on the theory of the engine with nonconducting walls. The theoretical engine carries on its expansion without the loss of any heat, while in the actual engine a large amount of heat is lost by radiation. There is also a considerable loss of pressure between the boiler and engine, due to resistance of flow through pipes and passages. In a slow-speed engine with large and direct ports and valves this trouble may be

minimized. The imperfect action of valve gears may also be lessened with due care, but the action of the cylinder walls still remains to be overcome.

In the theoretical card, admission is at constant boiler pressure, cut-off is sharp and expansion complete, that is, expansion continues until the temperature falls to that of the condenser and the exhaust is at condenser pressure. The piston also sweeps the full length of the cylinder.

In the actual engine there is a considerable loss of pressure between boiler and engine, and the wire-drawing of the ports and valves tends to cause a sloping steam line. Condensation at the beginning of the stroke causes the real expansion line to fall below the theoretical, while re-evaporation causes it to rise above the theoretical toward the end of expansion. In the actual

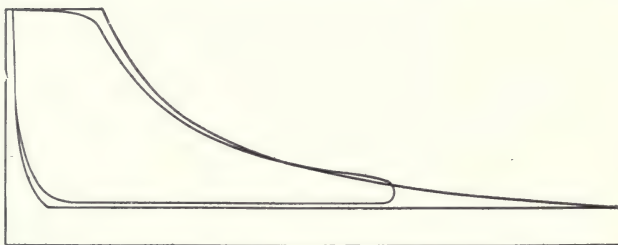


Fig. 4.

engine, release takes place before the end of the stroke, expansion is not complete, that is, the pressure at release is above that of the condenser, and the resistance of exhaust ports causes the back pressure to be above the actual condenser pressure. Moreover, the piston does not sweep the full length of the cylinder, and the clearance space must be filled with steam, which does little or no work. The theoretical and actual cards are shown in Fig. 4.

EFFICIENCY OF THE ACTUAL ENGINE.

We have seen that the efficiency of the theoretical engine is purely a thermal consideration ; the efficiency of the actual engine,

however, is a mechanical matter. The measure of the activity of work is the horse-power which corresponds to the development of 33,000 foot-pounds per minute. As 778 foot-pounds are equivalent to one B. T. U., 33,000 foot-pounds, or one horse-power, is equivalent to $33,000 \div 778 = 42.42$ B. T. U. Now if a certain engine uses 84.84 B. T. U. per horse-power per minute, it is evident that its efficiency would only be $\frac{1}{2}$ or 50 per cent, because $42.42 \div 84.84 = \frac{1}{2}$. Hence, we may say that the efficiency of

the actual engine is equal to $\frac{42.42}{\text{B. T. U. per H. P. per minute}}$. This

efficiency is always much less than that of the perfect engine. Let us now discuss the effects of some of the losses.

In the first place, the metal, being a good conductor of heat, becomes heated by the steam within and transmits this heat by conduction and radiation to the air or external bodies. With the cylinder well lagged much less heat is lost by radiation. If the lagging were perfect and the temperature of the cylinder remained the same as the temperature of the steam throughout the stroke, there would be no loss by radiation, but we should still lose heat by conduction to the different parts of the engine.

During expansion, the temperature and pressure of the steam decrease as the volume increases, and the temperature at exhaust is much less than the temperature at admission. In the perfect engine the working substance after exhaust is compressed to the temperature at admission, but in the actual engine much of this steam is lost and the compression of a part of it is incomplete, so that its temperature is less than the temperature at admission.

Suppose an engine is running with admission at 100 pounds absolute and exhaust at 18 pounds absolute. Then from steam tables we find the temperature at admission to be 327.6° , and at exhaust 222.4° . The metal walls of the cylinder, being good conductors and radiators of heat, are cooled by the low temperature of exhaust, so that the entering steam comes through ports and into a cylinder that is more than 100° cooler than the steam. This means that heat must flow from steam to metal until both are of the same temperature. This causes the steam to give up part of its latent heat, and as saturated steam cannot lose any of its heat without condensation, we find the cylinder walls covered with a

film of moisture known as initial condensation. This condensation in simple unjacketed engines working under fair conditions may easily be 25 per cent or more of the entering steam. The moisture in the cylinder has of course the same temperature as the steam; it has simply lost its heat of vaporization.

Although metal is a good conductor of heat it cannot give up nor absorb heat instantly; consequently during expansion the temperature of the steam falls more rapidly than that of the cylinder. This allows heat to flow from the cylinder walls to the moisture on them. As fast as the steam expands so that the pressure in the cylinder becomes less, this condensation will begin to evaporate. As the pressure falls it requires less and less heat to form steam, and therefore more and more of this moisture will be evaporated. At release the pressure drops suddenly, and more heat at once flows from the cylinder walls, and re-evaporation continues throughout the exhaust. Probably all of the water remaining in the cylinder at release is now re-evaporated, blows out into the air or the condenser, and is lost so far as useful work is concerned.

The steam that is first condensed in the cylinder does no work; its heat is used to warm up the cylinder, and later, when it is re-evaporated, it works only during a part of the expansion and at a reduced efficiency, because it is re-evaporated at a pressure and consequently at a temperature very much lower than that of admission. If the cut-off is short, perhaps 20 per cent of the steam condensed may be re-evaporated during expansion; if the cut-off is long, 10 per cent may be re-evaporated, the rest remaining in the cylinder at release still in the form of moisture. Thus some of the entering steam passes through the cylinder as moisture, until after cut-off, and still more passes clear through without doing any work at all.

Suppose an engine is using 30 pounds of steam per horse-power per hour and admission is at 100 pounds absolute. The latent heat of vaporization at this pressure is 884 B. T. U. per pound. If the condensation amounts to $33\frac{1}{3}$ per cent, then 10 pounds are condensed and we lose 10 times 884, equals 8,840 B. T. U. per hour, or 147.3 per minute; and since 42.42 B. T. U. represents 1 horse-power, we shall lose by condensation 147.3

divided by 42.42, equals $3\frac{1}{2}$ horse-power (nearly). If the cut-off is shortened, the condensation increases and may amount to 50 per cent at very short cut-off. Of course we use very much less steam at short cut-off than with long cut-off, and doubtless in many cases 50 per cent of the steam at short cut-off is not as great an absolute quantity as 30 per cent at long cut-off. Nevertheless, in all cases it is the percentages that go to make up the efficiency.

In addition to the actual loss from condensation in the cylinder there is still another loss due to the re-evaporation. Suppose, as before, that 10 pounds of steam are condensed in the cylinder, and that 20 per cent of this is re-evaporated during expansion. This will leave 8 pounds to be re-evaporated during exhaust. Suppose the exhaust is at 3 pounds above atmospheric pressure, or 18 pounds absolute (about). Then the heat of vaporization is 958.5 B. T. U. per pound of steam, and it will require 8 times 958.5, which equals 7668.0 B. T. U., to evaporate the 8 pounds. All of this heat is taken from the cylinder, leaving the engine much cooler than it would be were it not for this re-evaporation. This gives some idea of the great amount of heat passing away at exhaust, which is known as the exhaust waste.

In all cylinders it is necessary to have a little space between the cylinder cover and the piston when at the end of the stroke. In vertical engines the space is greater at the bottom than at the top. The volume of this space, together with the volume of the steam ports, is called the clearance. It varies from 1 to 20 per cent, depending upon the type and speed of the engine; the higher the speed, the greater the clearance. This clearance space must be filled with steam before the piston receives full pressure, but this steam does no work except in expanding, and the volume of the clearance offers additional surface for condensation.

Another important loss is that due to friction. We know that it takes considerable power to move an unloaded engine; if fitted with a plain, unbalanced slide valve, the power necessary to move the valve alone is considerable. The piston is made steam tight by packing rings, and leakage around the piston rod is prevented by stuffing boxes. All these devices cause friction as well as wear at the joints. The amount of power wasted in friction varies

greatly, depending upon the kind of valves, general workmanship, state of repair and lubrication.

MULTIPLE EXPANSION.

Two engines may be used together on the same shaft, partly expanding the steam in one of the cylinders, and then passing it over to the other to finish the expansion. One advantage from this arrangement is that the parts can be made lighter. The high-pressure cylinder can be of much less diameter than would be possible if the entire expansion were to take place in one cylinder. This, of course, makes the pressure exerted on the piston rod much less, and the piston rod and connecting rod can thus be made much lighter. The low-pressure cylinder must be larger than it otherwise would be, but its parts need not be much heavier, because the pressure per square inch is always low.

This arrangement gives not only the advantage of lighter parts, but a decided increase of economy over the single cylinder type. If attention is given to the matter, a loss of economy would be expected, because the steam is exposed to a much larger surface through which to lose heat, but the gain comes from another source, and is sufficient to entirely counterbalance the effect of a larger cylinder surface.

When very high pressure steam and a great ratio of expansion is used, the difference between the temperature of the entering and that of the exhaust steam is great. For instance, suppose steam at 160 pounds (gage) pressure enters the cylinder and the exhaust pressure is 2 pounds (gage), the difference in temperature is, from steam tables,

$$370.5^{\circ} - 218.1^{\circ} = 152^{\circ}.4$$

This difference becomes nearly 230° if the steam is condensed to about three pounds absolute pressure. The cylinder and ports of the engine are cooled to the low temperature of the exhaust steam, and, as we have seen, a considerable quantity of the entering steam is condensed to give up heat enough to raise the temperature of the cylinder to that of the entering steam. As the ratio of expansion increases, the difference in temperature increases, and consequently the amount of steam thus condensed also increases. To keep this initial condensation

as small as possible we must limit the temperature, that is, we must not have as great a difference between admission and exhaust. To do this we must divide the expansion between two or more cylinders.

It will be remembered that the great trouble Watt found with Newcomen's engine was its great amount of condensation, and he stated as the law which all engines should try to approach, "that the cylinder should be kept as hot as the steam which enters it." This is to avoid condensation when steam first comes in. If, instead of expanding the steam in one cylinder, we expand it partly in one and then finish the expansion in another, we shall have passed it out of the first cylinder before its temperature falls a great deal, and consequently the cylinder walls will be hotter than they would be if we had expanded it entirely in one cylinder. This would then reduce the amount of steam condensed. The importance of this may not be evident at first, but it makes a great difference in the economy of the engine. If there is less condensation, there will be less moisture to re-evaporate, and consequently less exhaust waste; hence we shall save in two ways at the same time.

In a compound engine we admit the steam first to the smaller or high-pressure cylinder, and exhaust it to the larger or low-pressure.

Suppose steam at 160 pounds (gage) pressure is admitted to a cylinder, and the ratio of expansion is such that the steam is exhausted at about 60 pounds (gage) pressure; then the difference of temperature is

$$370.5^{\circ} - 307^{\circ} = 63.5^{\circ}.$$

If, now, the steam when exhausted from the first cylinder enters a second and is allowed to complete its expansion, so that the exhaust pressure is about two pounds (gage) pressure, the difference of temperature in this cylinder will be

$$307^{\circ} - 218.5^{\circ} = 88.5^{\circ}.$$

Then for the single engine, if the exhaust pressure is two pounds (gage), the difference of temperature is 152° , while in the compound engine this difference is divided into two parts,

63.5° and 88.5°. The cylinder condensation for both cylinders of the compound engine will be much less than if the total expansion took place in a single cylinder. The cylinders should be so proportioned that the same quantity of work may be done in each.

If there are two stages of expansion, the engine is called compound, three stages triple, and four quadruple.

Besides reducing the excessive condensation, there is still another gain in using multiple expansion. We have seen how much heat is lost by the exhaust waste, which in the simple

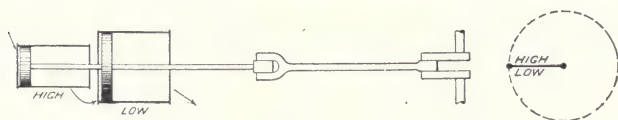


Fig. 5.

engine blows into the air or into the condenser, and is entirely lost. In the multiple expansion engine the exhaust and re-evaporation from one cylinder passes into the next and does work there; any

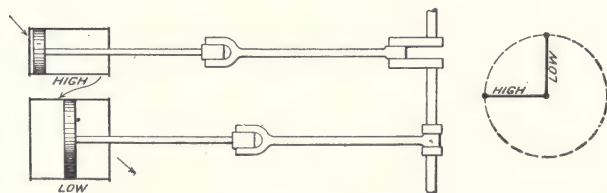
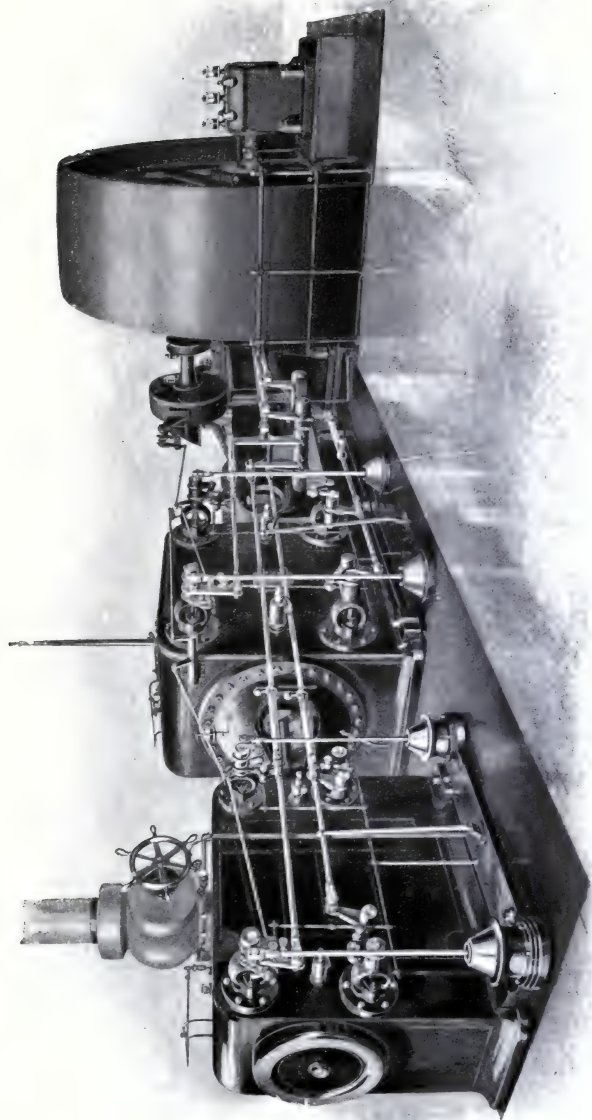


Fig. 6.

leakage from the high pressure is also saved, and does work in the low.

Mechanically there is a decided advantage; the several cranks give a more even turning moment, and the distribution of work between two or more cylinders makes it possible to use lighter individual parts; but there is a disadvantage in having more parts to look after, and a greater first cost of the engine.



HORIZONTAL TANDEM RICE AND SARGENT ENGINE.
Providence Engineering Works.

Engines having two or more cylinders are arranged in various ways, but only the most common methods will be shown here. For two-cylinder compound engines the cylinders are often placed tandem, as shown in Fig. 5. This has the advantage of having but one crank, connecting rod and crosshead, but it has the disadvantage of dead points. In other words, the turning moment on the shaft is no more uniform than for a simple engine. When the cylinders are placed side by side and the cranks are at right

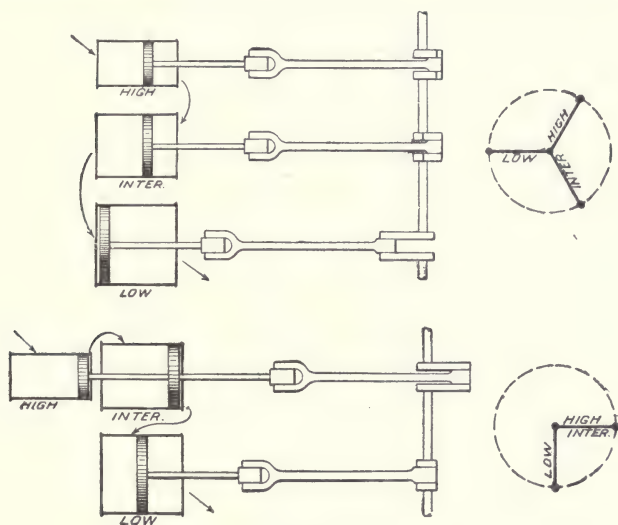


Fig. 7.

angles, as shown in Fig. 6, the low-pressure cylinder cannot receive the exhaust steam directly from the high-pressure cylinder; consequently a receiver must be used. In this case the advantage is due to the even turning moment, and the disadvantage is from the cost of receiver, extra crosshead, crank and connecting rod. In many instances, as electric lighting, marine work, etc., the advantage of the more nearly constant force acting on the shaft is worth far more than the extra first cost. The same may be said

of triple and quadruple expansion engines. (See Fig. 7.) Each type of engine has its special service, and what is suitable in one case may not be serviceable in another. Fig. 8 shows the diagram of a quadruple expansion engine, which is more commonly used in marine work.

JACKETING.

Another method of reducing the loss due to cylinder condensation is to supply heat to the steam while it is in the cylinder.

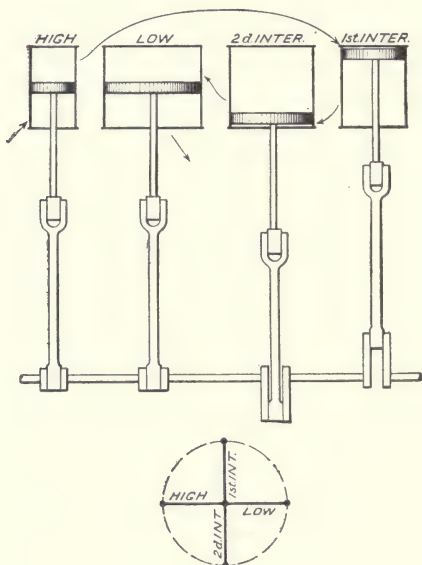


Fig. 8.

This is done by surrounding the cylinder with an iron casting and allowing live steam to circulate in the annular space thus formed. The cylinder covers are also made hollow to permit a circulation of live steam. A cylinder having the annular space (A, Fig. 9) filled with steam is said to be jacketed. A liner, L, is often used in jacketed cylinders.

The function of the jacket is to supply heat to the cylinder walls, to make up for that abstracted during expansion and exhaust, so that at admission the cylinder will be as hot as possible. The result is, that the difference in temperature between the cylinder walls and the entering steam is considerably less than in engines where no jacket is used. Condensation is therefore reduced, and since heat flows from the jacket to the cylinder during expansion, a much larger amount of this condensation is re-evaporated before release and it thus has a chance to do some work in the cylinder. This leaves a comparatively small amount of exhaust waste, and the heat thus abstracted is made good from the steam

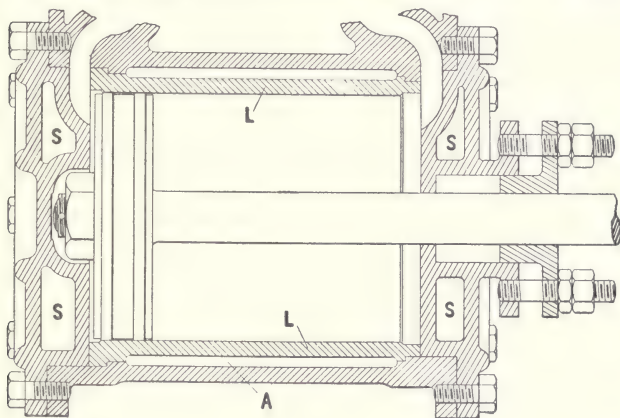


Fig. 9.

in the jacket. Since a large amount of heat is given up by the jacket steam a good deal of it must be condensed. Thus the question is asked: 'What is the advantage of this method over that of allowing the entering steam to supply the heat by its own condensation?' This question is answered briefly as follows:

The loss of heat by condensing the steam would be less if the inside of the cylinder could be kept dry. We have seen how the moisture that collects by condensation is re-evaporated during expansion and exhaust, because the pressure falls and the cylinder walls are hotter than the steam. This re-evaporation takes place at the

expense of the heat in the cylinder walls, and they are thus cooled. We have already seen that a great many B. T. U.'s are thus taken from the cylinder and thrown out at exhaust at every stroke. Now if we can keep the inside dry, so that there will be little or no re-evaporation at exhaust, we shall make a considerable saving. The steam that condenses in the jacket does not re-evaporate; it is returned to the boiler as feed water, so that the only heat lost is the latent heat given up during condensation. If the cylinder is heated from within, both the latent heat given up by condensation and the latent heat required for re-evaporation are lost.

In a triple expansion engine there is one distinct advantage in allowing condensation in the cylinder, for this moisture acts as a lubricant, and as the heat of re-evaporation passes into the next cylinder and there does work, there is very little loss. In many marine engines there are no means of lubricating the cylinders at all, the engineers depending entirely upon the condensation.

SUPERHEATED STEAM.

We have already learned that superheated steam contains more heat than is necessary to keep it in the form of steam, and that consequently it can part with some of that heat without condensing.

Suppose this superheated steam is admitted to the cylinder of an engine. We know that the cylinder walls would be comparatively cool, as they were in contact with the exhaust steam and were still further cooled by the re-evaporation of water which was condensed on them at the previous admission of steam. Heat must flow from the steam to the walls until they are as hot as the steam. If the steam were merely saturated, as is usually the case, some of it would be condensed in order to supply this heat to the walls. But in the case of the superheated steam we know heat was given to it beyond that necessary simply to make saturated steam, so that it can give up some heat to the walls before it begins to condense.

Thus the amount of initial condensation is materially reduced, and consequently there will be less cooling of the cylinder walls by the re-evaporation of this condensation; therefore the walls will be hotter at the next admission, and it will require less heat to raise them to the temperature of the incoming steam.

Besides the saving in the engine there is a considerable gain due to the increased efficiency of the boiler, for by superheating, a part of the heat in the waste gases is utilized. If a boiler primes or is so constructed that it does not furnish dry steam, the gain from superheating will be most marked. If there is moisture in the steam and this passes over into the engine, the heat that it contains is entirely wasted, and the re-evaporation of this additional moisture during exhaust causes the cylinder walls to cool off still further.

Suppose we have steam in a boiler at 60 pounds pressure by the gage. Its temperature will be about 307° F. If this steam leaves the boiler by a pipe which passes through a furnace or chimney where the temperature is greater than 307° , it will become heated as it passes through. The pressure will, however, be the same, for it is still in communication with the boiler. This is actually the way in which steam is superheated in practical work.

One great difficulty in superheating is that the superheater, which is usually placed in the uptake, is subjected to a very high temperature and there is great danger of its burning out. The plates of a boiler are not likely to be damaged by the intense heat of the furnace, because the water by its good conducting power prevents their becoming overheated. But steam is a poor conductor of heat, and although the superheater is placed in the uptake or chimney, where the heat is not as intense, they nevertheless cause a great deal of trouble.

Superheating is efficient if low pressure is used, that is, steam at about 50 pounds pressure; but if saturated steam is at 150 to 200 pounds pressure, the temperature is high. If more heat is then added, the temperature becomes such that lubricants are decomposed. In such cases the piston and valve, not being lubricated, cut into the cylinder walls and the valve seat, thus causing leaks and requiring excessive power to move the engine. The packing in the stuffing boxes, unless metallic, is injured and causes considerable trouble. In case high-pressure steam is used the superheating must be slight, often so slight as not to be worth the trouble and expense.

CONDENSERS.

In our study of the theoretical engine on page 20, we learned that to meet the ideal conditions and to attain the maximum efficiency, steam must enter the engine at the constant temperature of the heat generator (or boiler) and must leave the engine at the constant temperature of the refrigerator (or condenser). We have also learned that the difference between the temperature of admission and exhaust is a measure of the thermal efficiency. In Part I of "The Steam Engine," Watt's principle was stated, namely, that "when steam was condensed it should be cooled to as low a temperature as possible." In our discussion of saturated vapors we also learned that the temperature varied with the pressure, and consequently to cool the condensed steam to as low a temperature as possible simply means to condense the steam to as low a pressure as possible.

In the ordinary noncondensing engine, steam cannot be expanded below a pressure of 14.7 pounds, because the atmosphere exerts that pressure at the opening of the exhaust pipe. In fact, this 14.7 pounds is only the theoretical limit, and in practice the exhaust is always a little above this because of resistance in the exhaust ports and exhaust pipe; 17 or even 18 pounds back pressure is more nearly the conditions of actual service.

During the forward stroke, steam expands from the pressure at admission to a much lower pressure at release; then the valve opens for the return stroke and on one side of the piston there is full steam pressure, and on the other side the pressure of exhaust, which acts against the piston and against the force of the incoming steam. If all of this back pressure could be removed so that there would be a vacuum on the exhaust side of the piston, the power of the engine would be increased by just so many pounds of M. E. P., and in addition to this the steam could expand to a very much lower pressure and therefore work with greater economy.

One pound of steam at 17 pounds absolute pressure occupies 23.22 cubic feet of space in the cylinder of the engine, but one pound of water in the condenser occupies only about 0.016 cubic foot, which makes the steam occupy nearly 1,450 times as much space as the water into which it condenses. If, then,

the exhaust steam could be condensed instantly, the back pressure would be reduced almost to zero and the engine would exhaust into a vacuum.

We know that a certain amount of heat is required to change one pound of water at a given temperature into steam at the same temperature; this is called the latent heat, or heat of vaporization. If the steam condenses, it must give up this latent heat. The easiest way of doing this is to let the steam mingle with a spray of water, as in the jet condenser, or come in contact with pipes through which cold water is circulated, as in the surface condenser. These two forms of condenser are fully described in Part I and will not be further considered here.

Unfortunately the mere condensation of the steam will not give a perfect vacuum, because more or less air, which is always in the water, comes over from the boiler and thus gets into the condenser. Moreover, the condensed water is hot, and a vapor rises from it in the condensing chamber; this, together with the air and some leakage, would spoil the vacuum were it not for the air pump, which removes the air and condensed steam. With the best air pump it would be impossible to maintain a perfect vacuum, but a vacuum of 26 inches, which corresponds to about 2 pounds absolute pressure, can readily be maintained in good practice.

Advantages of Condensing. It has already been stated that there is a gain in thermal efficiency by running an engine condensing, but it will be more clearly seen by considering a few figures. The thermal efficiency may be expressed by the formula:

$$E = \frac{T_1 - T_2}{T_1}.$$

This efficiency may be increased by making T_1 larger, which would happen if the boiler pressure were increased, or by making T_2 smaller, which would result from reducing the back pressure by condensing. If the boiler pressure is raised, both the numerator and denominator of the fraction will increase, and the value of the fraction will be but slightly greater. If, however, the back pressure is reduced, the numerator, $T_1 - T_2$, will be larger, while the denominator, T_1 , will remain the same. It is apparent that this will cause a much greater increase in efficiency than raising the boiler pressure a like amount.

Suppose an engine is supplied with steam at 85.3 pounds (gage) pressure and it exhausts at 3.3 pounds (gage) pressure. The absolute temperature corresponding to $85.3 + 14.7 = 100$ pounds pressure is $327.58 + 461 = 788.58$, and the absolute temperature corresponding to $3.3 + 14.7 = 18$ pounds pressure is $222.40 + 461 = 683.40$. Then the thermal efficiency will be from the formula:

$$\frac{T_1 - T_2}{T_1} = \frac{788.58 - 683.40}{788.58} = .133 \text{ or } 13.3 \text{ per cent.}$$

If the boiler pressure were raised to 140 pounds absolute the efficiency would be

$$\frac{813.85 - 683.40}{813.85} = .16 \text{ or } 16 \text{ per cent.}$$

If instead of increasing the boiler pressure a condenser is used and thereby the exhaust pressure reduced to 4 pounds (absolute), the efficiency becomes

$$\frac{788.58 - 614.09}{788.58} = .221 = 22.1 \text{ per cent.}$$

Thus we see that if we lower the exhaust pressure 14 pounds we get a greater increase in efficiency than if the boiler pressure is raised 40 pounds.

Another method of showing the advantage by condensing the exhaust is by the indicator card.

Fig. 10 represents a card from a $12'' \times 20''$ engine making 75 revolutions with 75 pounds steam pressure. The dotted diagram represents a card taken when running without a condenser. Cut-off occurs at $\frac{1}{3}$ the stroke. The M. E. P. is 44.2 pounds. Hence the indicated horse-power is about 37.87.

The card shown in full line was taken with the same load, same speed, and with a condenser producing a vacuum of about 26 inches of mercury, which is equal to about 12.7 pounds. The absolute pressure at exhaust is then $14.7 - 12.7 = 2$ pounds. Since the load is the same, the areas and the M. E. P. must be the same in both cases. Cut-off is found to be only about $\frac{1}{6}$ the stroke. The above engine without the condenser uses an amount of steam represented by the length A C, while if a condenser is attached the steam used is represented by the length A C'. Thus

we see that the amount of steam consumed per stroke is considerably less if a condenser is used.

Quantity of Water. Besides merely condensing the steam the injection water cools it still further, so that more than merely the latent heat is removed from it. If exhaust steam enters the condenser at a temperature t_1 , it contains a certain amount of heat, which is the *total heat* at that temperature. If it is condensed and cooled to a temperature t_2 , at which it leaves the condenser, it then contains a certain amount of heat which is the *heat of the liquid* at this temperature t_2 .

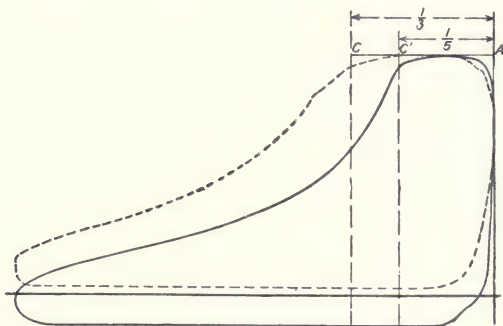


Fig. 10.

If A represents the total heat at t_1 , and B represents the heat of the liquid at t_2 , then the heat given up by one pound of condensed steam is equal to $(A - B)$ B. T. U., provided the exhaust that enters the condenser is dry saturated steam. If C is the temperature of the injection water, and D is the temperature of the discharge water, then every pound of cooling water absorbs one B. T. U. for every degree rise in temperature; or we may say that the heat absorbed is equal to $(D - C)$ B. T. U. per pound of cooling water. Then it will take as many pounds of water to absorb $(A - B)$ heat units as $(D - C)$ is contained times in $(A - B)$. We may express this in terms of a formula thus:

$$W = \frac{A - B}{D - C}$$

In which W = pounds of cooling water per pound of steam.

For example: Suppose steam is expanded in an engine to 4 pounds absolute pressure. If the temperature of the injection water is 45°, and the condenser is of the surface type with discharge water at 120°, and the temperature of the condensed steam is 130°, how many pounds of injection water are required per pound of steam?

By consulting the steam tables we find the total heat of steam at 4 pounds pressure to be 1,128.6 B. T. U. The heat of the liquid in the condensed steam at 130° is 98.1 B. T. U. Then

$$W = \frac{1,128.6 - 98.1}{120 - 45} = 13.74 \text{ pounds.}$$

Suppose steam at 6 pounds absolute pressure exhausts into a jet condenser. The temperature of the injection water is 50° and the discharge is 120°. How many pounds of water are necessary to condense 8 pounds of steam?

In the jet condenser the temperature of the condensed steam and discharge water is the same. We find from the steam tables that the total heat of steam at 6 pounds absolute is 1,133.8 B. T. U., and the heat of the liquid in the condensed steam at 120° is 88.1 B. T. U. Then, as before,

$$W = \frac{1,133.8 - 88.1}{120 - 50} = 14.94.$$

For 8 pounds it will take $14.94 \times 8 = 119.52$ pounds.

The above calculation cannot be relied upon to any great extent, for we seldom know the true conditions in the condenser, and it would be of little value to us if we did know, as the exact conditions will change considerably. In practice it is customary to allow for about twice as much water as the above calculation would require. These figures give us a fair idea of the necessary sizes of pipes and passages leading to the condenser, and give a basis for estimating the dimensions of the air pump.

Cooling Surface. The amount of surface required to condense the steam in surface condensers depends upon the efficiency of the metal, the condition of the tubes, the difference in temperature between the two sides and their thickness. There have been no satisfactory tests to determine the amount of cooling surface

necessary for a condenser, and in actual practice there seems to be a wide diversity of opinion. The tubes of a condenser are much thinner than boiler tubes, and much more clean, hence we might expect them to be more efficient in condensing the steam than the boiler tubes are in evaporating it. Boilers may evaporate 7 pounds or more of steam per hour, per square foot of heating surface. Seaton, an eminent authority on marine work, says that with cooling water at 60° F, and the discharge at 120°, a condensation of 13 pounds of steam per square foot per hour is fair average work. A new condenser will of course condense much more than this. If the exhaust pressure is from 6 pounds to 10 pounds absolute, an allowance of 1.5 to 1.8 square feet of cooling surface may be allowed per indicated horse-power, depending upon the pressure. This assumes that the temperatures of the injection and discharge water shall be 60° and 120° respectively.

It is evident that the amount of surface will depend upon the quantity of steam used per hour by the engine, the pressure and temperature of the exhaust and the temperature of the cooling water and discharge. There must also be an allowance for inefficient work after the condenser has become fouled with service. All these conditions make the problem so uncertain that calculations by means of formulæ are likely to be untrustworthy, and it is best at all times to make estimates from the figures given for similar conditions in actual service.

Measurement of Vacuum. We have seen that in order to maintain a vacuum in the condenser it is necessary to pump out, by means of an air pump, the air that leaks in. Evidently, if we are to maintain a proper vacuum, it is necessary to know at all times just how much pressure there is in the condenser. If the pressure increases, the air pump can be run a little faster until the proper vacuum is obtained.

From the study of pneumatics we know that the pressure of the atmosphere can be measured by means of a column of mercury. The atmospheric pressure will support a column of mercury about 30 inches high, which is equivalent to a pressure of 14.7 pounds (nearly). It would be inconvenient to attach a mercury column to the condenser and so we use a gage, in general appearance similar to a boiler gage except that the dial is graduated to read

in inches of mercury instead of in pounds pressure, and it indicates the inches of vacuum *below atmospheric pressure*. If the pointer of the vacuum gage stands at 20, it means that the pressure is equal to 20 inches below atmospheric pressure. Since 30 inches is equal to 14.7 pounds, 20 inches would be equal to $\frac{20}{30} \times 14.7 = 9.8$ pounds.

This is 9.8 pounds below atmosphere, or $14.7 - 9.8 = 4.9$ pounds above zero.

If the vacuum gage stands at 26 inches, what is the absolute pressure in the condenser?

$$\frac{26}{30} \times 14.7 = 12.74 \text{ pounds below atmosphere}$$

$$14.7 - 12.74 = 1.96 \text{ pounds absolute pressure.}$$

EXAMPLES FOR PRACTICE.

1. Steam enters the condenser at 11 pounds pressure (absolute). The water enters at 53° and leaves at 110° . The condensed steam leaves at 120° . If the engine uses 26 pounds of steam per horse-power per hour while running at 53 horse-power, what would be the theoretical amount of water used by the condenser and what should be the area of cooling surface of the condenser?

$$\text{Ans. } \begin{cases} 106 \text{ square feet.} \\ 25,479 \text{ pounds.} \end{cases}$$

2. Steam enters a jet condenser at 16 pounds absolute pressure. The injection water has a temperature of 48° . If the temperature of the discharge water is 115° , how many pounds of injection water are necessary per pound of steam?

$$\text{Ans. } 15.89 \text{ pounds.}$$

3. If the pointer of a vacuum gage stands at 9, what is the approximate absolute pressure in the condenser?

$$\text{Ans. } 10.3 \text{ pounds.}$$

4. The pointer of a vacuum gage which is attached to a condenser stands at 23. What is the approximate pressure in the condenser?

5. The steam gage indicates 175 pounds and the vacuum gage 26 inches of mercury. What is the total difference in pressure?

$$\text{Ans. } 187.74 \text{ pounds.}$$

OTHER DEVICES.

Corliss Valves. From a thermal point of view the advantages of the Corliss valve may be summed up in a few words. The exhaust valves are separate from the admission valves and hence the exhaust steam does not come in contact with the admission ports. Thus the admission ports are not cooled and there is less condensation in proportion to the ratio of expansion than in the plain slide valve type of engine. The short ports reduce the volume of clearance and thus save clearance steam and reduce the surface exposed to condensation.

Separator. We have studied the loss of heat due to re-evaporation at exhaust and find that it is considerable. The more moisture there is in the entering steam, the more moisture there will be to re-evaporate; consequently more heat will be lost from the cylinder. It will at once be seen that if we can separate the moisture from the entering steam, we shall keep considerable water out of the cylinder.

PISTON SPEED.

The total distance passed over by the piston in one minute is called the piston speed. As the piston travels the length of the stroke twice for every revolution, the piston speed is evidently equal to the length of stroke multiplied by twice the number of revolutions. Then in the formula for horse-power, $\frac{PLAN}{33,000}$,

N = the number of strokes

$L \times N$ = the piston speed.

In case the stroke is stated in inches, the piston speed in feet equals $\frac{L \times N}{12}$.

In determining the allowable piston speed, local conveniences, durability and the character of the work to be done should be considered. Other things being equal, the piston speed increases slightly with an increase in the length of stroke.

The piston speeds in common practice are as follows:

Direct acting pumping engines	100 to 150 feet per minute
Small stationary engines	300 to 500 feet per minute
Large stationary engines	500 to 900 feet per minute
Corliss engines	400 to 800 feet per minute
Locomotives	700 to 1,200 feet per minute
Marine engines	750 to 1,000 feet per minute

RATIO OF EXPANSION.

The ratio of expansion as usually understood is the piston displacement divided by the volume of the cylinder to the point of cut-off. Thus if the piston is 10 inches in diameter and the stroke is 12 inches, the piston displacement is $10^2 \times .7854 \times 12 = 942.48$ cubic inches. If cut-off occurs at $\frac{1}{3}$ the stroke, or when the piston has moved 4 inches, the volume to cut-off is $10^2 \times .7854 \times 4 = 314.16$ cubic inches, and the ratio of expansion is $\frac{942.48}{314.16} = 3$. This may be stated thus:

$$\frac{\text{area of cylinder} \times \text{length of stroke}}{\text{area of cylinder} \times \text{distance to cut-off}} = \text{ratio of expansion.}$$

Since the area of the cylinder appears in both numerator and denominator, we may cancel and write:

$$\text{Ratio of expansion} = \frac{\text{length of stroke}}{\text{distance to cut-off}}.$$

If clearance is taken into account, the true ratio of expansion is different from the apparent ratio as found above. The steam which expands in the cylinder is not merely the volume equal to the piston displacement, but is this volume plus the volume of the clearance.

With a late cut-off and small clearance the true ratio of expansion differs but little from the apparent. But when the cut-off is early and the clearance is large there is considerable difference, as shown in Fig. 11.

In this diagram D-E represents the stroke, and is $2\frac{1}{2}$ inches long. D'D represents the clearance volume, which is equal to $\frac{1}{3}$ the piston displacement, and is therefore .31 inch long. Let cut-off occur at $\frac{1}{3}$ of the stroke; then the apparent ratio of expansion is equal to $\frac{D'E}{A'C} = \frac{2\frac{1}{2}}{.31} = 8$. If we consider the clearance, the

diagram is changed, as shown by the dotted lines, and the true ratio of expansion is

$$\frac{D'E}{A'C} = \frac{2.86}{.62} = 4.5.$$

Thus we see that the ratio of expansion is not 8 but 4.5.

WORK DONE IN THE CYLINDER.

The amount of work done per stroke in the engine cylinder is usually found by means of an indicator. It is often desirable, especially in designing engines, to know the work that may be expected under given conditions before the engine is built. The work done in the cylinder is proportional to the mean effective pressure, so that if we are to determine the probable power of our engine we must in some way ascertain the probable mean effective pressure. If the initial pressure, the back pressure and the ratio of expansion are known, we can find the probable M. E. P. by

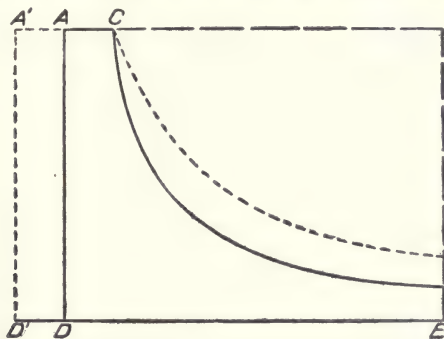


Fig. 11.

means of a formula. This formula does not provide for the losses due to compression, clearance, condensation and leakage, so that the value thus obtained must be multiplied by a factor depending upon the type of engine.

Let P = absolute initial pressure

R = ratio of expansion

p = back pressure (absolute)

Then neglecting all losses, the theoretical M. E. P. will be expressed by the formula,

$$\text{M. E. P.} = P \frac{(1 + \text{hyp. log. } R)}{R} - p.$$

In order to allow for the losses due to compression, clearance,

condensation, wire-drawing, etc., it is customary to multiply this value by factors given in the following table :

Engines with special valve gear and independent cut-off valves, about	.90
Plain slide-valve engines	.75 to .90
Compound engines	.65 to .80
Triple expansion engines	.50 to .70

These factors are for unjacketed engines ; with jackets the factor might be .05 more in each case. On account of wire-drawing and large compression, high-speed engines would have the smaller value in its respective class. Only the best-designed engines with good valve gears and under favorable conditions will warrant the use of the larger factor in their respective classes. Large engines will of course have larger factors than small engines.

Example.—What is the probable M. E. P. when the initial pressure is 95.3 pounds (gage), the back pressure 3 pounds (absolute) and the ratio of expansion 4? The engine is large, of the plain slide-valve type with steam jackets.

$$P = 95.3 + 14.7 = 110 \text{ pounds}$$

$$p = 3 \text{ pounds}$$

$$R = 4$$

From the following table, hyp. log. $R = 1.3863$

$$\text{M. E. P.} = \frac{110 (1 + 1.3863)}{4} - 3$$

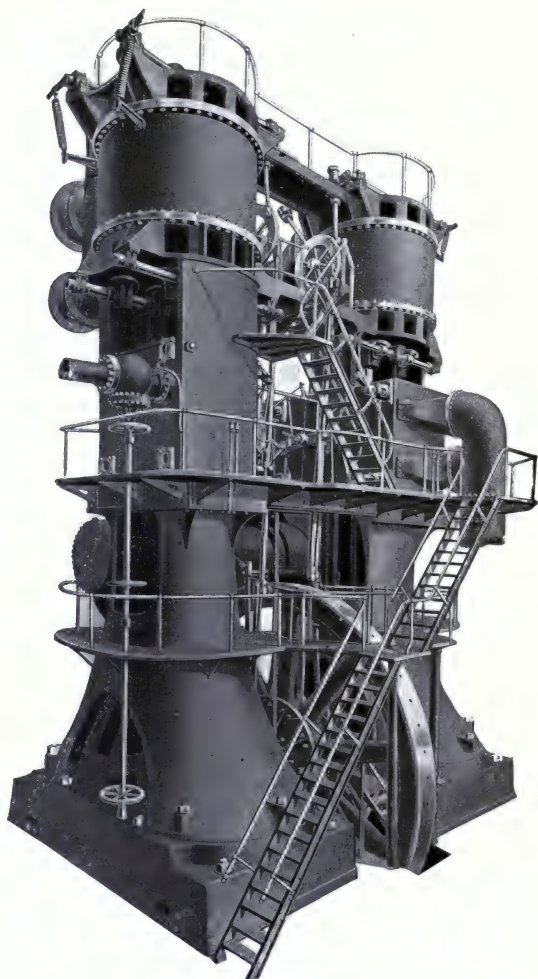
$$= 62.62 \text{ pounds.}$$

For a large plain slide-valve engine, unjacketed, we might expect a factor of .85 to .90, depending upon the conditions. Suppose we call the factor .87, then adding .05 for the effect of the steam jacket we get .92.

$$62.62 \times .92 = 57.6 \text{ pounds as the probable M. E. P.}$$

A rough approximation to the power to be expected of a compound, or triple expansion, engine, may be found in a similar manner by assuming that all the expansion takes place in the low-pressure cylinder, neglecting losses as before.

Example.—Find the probable I. H. P. for a triple expansion engine with the following dimensions and data :



BLOWING ENGINE.

Steam Cylinder $58 \times 108 \times 60$ inches. Air Cylinders 108 inches diameter

Total Weight of Engine 1,270,000 Pounds.

William Tod Company.

Revolutions,	115 per minute
Initial pressure,	140.3 pounds (gage)
Back pressure,	2 pounds (absolute)
Total ratio of expansion,	12
Diameter of low-pressure cylinder,	60 inches
Stroke,	30 inches

$$\begin{aligned}
 \text{M. E. P.} &= \frac{P(1 + \text{hyp. log. } R)}{R} - p \\
 &= 155 \frac{(1 + 2.48490)}{12} - 2 \\
 &= 43.01.
 \end{aligned}$$

In practice only about .65 of this is actually obtained, so the probable M. E. P. is $.65 \times 43.01 = 27.95$ pounds.

The I. H. P. is now found from the expression $\frac{\text{PLAN}}{33,000}$ as follows:

$$\text{I. H. P.} = \frac{27.95 \times 2\frac{1}{2} \times 2,827.4 \times 2 \times 115}{33,000} = 1,377.$$

TABLE OF HYPERBOLIC OR NAPIERIAN LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.25	.22314	3.75	1.32175	6.25	1.83258	8.75	2.16905
1.50	.40546	4.00	1.38629	6.50	1.87180	9.00	2.19722
1.75	.55962	4.25	1.44692	6.75	1.90954	9.25	2.22462
2.00	.69315	4.50	1.50407	7.00	1.94591	9.50	2.25129
2.25	.81093	4.75	1.55814	7.25	1.98100	9.75	2.27726
2.50	.91629	5.00	1.60943	7.50	2.01490	10.00	2.30258
2.75	1.01160	5.25	1.65822	7.75	2.04769	12.00	2.48490
3.00	1.09861	5.50	1.70474	8.00	2.07944	15.00	2.70805
3.25	1.17865	5.75	1.74919	8.25	2.11021	18.00	2.89038
3.50	1.25276	6.00	1.79176	8.50	2.14006	20.00	2.99570

CRANK ACTION.

In the steam engine the steam exerts a pressure on the crank pin through the piston rod and connecting rod. When the crank is at the dead center, the entire pressure is on the bearing of the crank shaft; there is no tendency to turn the crank. In any other position the steam pressure tends to turn the crank pin. As the crank pin moves from dead center, the tendency increases until it

reaches a maximum and then decreases until, at the other dead center, it is zero again. If the connecting rod were of infinite length, and steam were admitted throughout the whole stroke, the maximum tendency, or the maximum turning moment as it is called, would occur with the crank at right angles to the line connecting the dead points.

In the actual engine the thrust along the rod is constantly varying even though the pressure on the piston remains the same. This is due to the angularity of the connecting rod. The turning moment is always equal to the thrust along the connecting rod multiplied by the perpendicular distance from the connecting rod to the center of the shaft. If the steam pressure on the piston remains constant, the maximum turning moment occurs

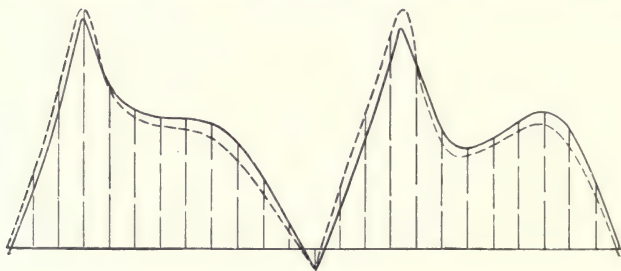


Fig. 12.

when the connecting rod is at right angles to the crank, for in this position the perpendicular distance from rod to center of shaft is a maximum equal to the length of the crank, and as the rod makes its greatest angle with the line connecting the dead center at this point the thrust along it will also be a maximum. If the cut-off is very early, $\frac{1}{4}$ stroke for instance, the maximum thrust along the rod will occur earlier than at the point previously mentioned, but the leverage of the force will be less, so that really there will be little change in the point of maximum turning moment no matter where the cut-off may occur.

To represent this turning moment, diagrams of crank effort are drawn. These diagrams may be drawn with rectangular co-ordinates having the crank angles represented as abscissae and

the turning moments corresponding to these angles as ordinates.

Besides the thrust of the connecting rod we must take into account friction and the inertia of the reciprocating parts. At first this may be thought of small consequence, but with a fairly heavy piston and connecting rod we can easily see that at high speed the momentum would be great. On a vertical engine, on the up stroke the steam has to lift this heavy mass and impart a very considerable velocity to it, while on the down stroke the acceleration of the mass is added to the steam pressure. This makes the effective force on the up stroke less than that due to the actual steam pressure, and greater on the down stroke. The crank effort diagram represented in Fig. 12 is from a horizontal engine of practical proportions. The initial steam pressure is 50 pounds per square inch. Cut-off at $\frac{1}{6}$ stroke. The engine makes

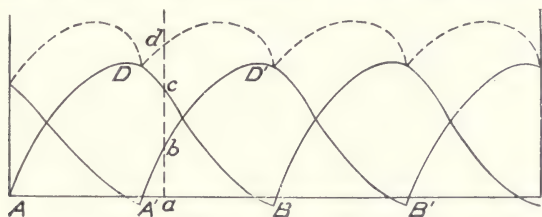


Fig. 13.

240 revolutions per minute. The dotted lines represent the crank effort, without considering friction. The full line is the diagram when weight of moving parts and inertia are considered.

In drawing a crank effort diagram, friction is often neglected, but the effect of inertia of the moving parts is of great importance, especially in the case of high speed.

It has been mentioned in connection with compounding that if there are two or more cranks on the shaft the turning moment is more nearly constant. We can now see that this is so. In Fig. 13, ADB represents the curve of turning moments on one crank, and A'D'B' the curve of turning moments on the other crank, which is at right angles to the first. To find the total turning moment on the shaft the dotted line curve is drawn.

The points are obtained by adding the ordinates. Thus, to find the point d add ac to ab . Then $ad = ac + ab$. It is easily seen that the dotted line curve is more nearly a straight line than the curve $A D B$.

ECONOMY OF THE STEAM ENGINE.

The effects of various devices and methods employed to increase the economy of the steam engine cannot be studied from the theoretical side alone if we wish to obtain satisfactory results. So complicated and important is the action of the cylinder walls, that if we would learn the conditions favorable to economy we must study the actual tests of engines in service conditions.

Effect of Raising Steam Pressure. In general there is a gain in thermal efficiency by increasing the steam pressure and the total number of expansions, provided proper means are taken to lessen the undesirable effects due to increased condensation and re-evaporation caused by the increased expansion. The initial condensation, of course, places a pretty strict limit upon the number of expansions profitably used in a simple expansion engine.

If we raise the steam pressure and keep the same cut-off, the conditions will be slightly different. The losses from external radiation increase with the rise of steam pressure, but the horsepower increases more rapidly, hence there is a net gain, and since the changes of temperature are greater, the higher the pressure, it is reasonable to suppose that the loss from condensation and re-evaporation will increase. It is not difficult to imagine a point where these losses may offset the gain, and where, indeed, if the pressure is raised too far there will be a net thermal loss. This has actually happened in some cases. At first the gain from raising the steam pressure is rapid. As the pressure rises, the gain increases more slowly, and finally it is not worth the expense, if, indeed, there is not actual loss.

If we bear in mind that there is little gain in economy to be obtained by increasing the steam pressure beyond a certain moderate limit, unless the ratio of expansion is also increased, and that the losses from condensation, re-evaporation and exhaust waste limit the number of expansions profitably used in a simple expansion engine, we shall at once see that we must look to other devices

for decided gains in economy. The most important of these have already been briefly considered.

Superheated Steam. Tests show a very decided gain in economy from using superheated steam, yet practically no permanent results have been obtained, on account of the difficulty of maintaining the superheating apparatus. To superheat to the best advantage we must have a coil of pipe in the flue and thus make use of the otherwise wasted heat. But the intense heat soon burns out the metal and together with the great pressure it is rapidly wasted away. Gains of from 15 per cent to 20 per cent from using superheated steam have actually been observed on simple engines. There has also been some gain noted by the use of superheated steam in triple expansion engines, but the gain is less than in the simple type, because naturally there is less condensation in the triple than in the simple engine.

Steam Jackets supply a small amount of heat to the cylinder during expansion which can be converted into work, but the chief advantage of a jacket is that it keeps the inner walls of the cylinder warmer at admission and thus reduces initial condensation and saves much loss of re-evaporation at exhaust. It will be evident that a large part of the heat of the steam jacket flows to the cylinder during exhaust and is thus entirely lost in the simple engine. In the triple engine this heat passes into the intermediate and low-pressure cylinders; consequently we might expect a greater gain from using a jacket on a triple than on a large simple engine. The main advantage of the jacket has been previously pointed out, and it may be stated that in all cases the gain from the jacket is small and there is to be found a considerable diversity of opinion as to its real advantages. On some engines there is undoubtedly little if any gain. The largest gain is in the smaller engines of say under 200 H. P.; on very small engines the gain is quite large, having been as much as 30 per cent on a 5" $\times$ 10" engine when developing only $1\frac{1}{2}$ H. P. under light load. On a 10 H. P. engine the gain might be as much as 25 per cent. On engines of about 200 H. P. the gain would probably be 5 per cent to 10 per cent for simple condensing and compound condensing, and from 10 per cent to 15 per cent for triple expansion. The saving on large engines, of say 1,000 H. P., is very small, the reason being

that large engines offer less cylinder surface per unit of volume than small ones, and hence we find proportionately less cylinder condensation in large engines than in small ones. The very small engines, in which the gain would be greatest, are seldom jacketed, because they are built for inexpensive machines and the first cost is of more consequence than the later economy.

Compounding is the most effective method of increasing the number of expansions and at the same time avoiding excessive cylinder condensation. We know that increase in boiler pressure and increase in expansion in simple engines are economical only to a certain limit. We shall now discuss the gain due to compounding and the conditions under which it is advantageous.

A direct comparison between tests of different engines is impossible because of the different steam pressures, etc., but a careful study seems to show that for simple condensing engines there is no advantage in raising the steam pressure above 80 pounds. In compounding, the pressure can be advantageously raised to 135 pounds. The gain due to the higher pressure, greater number of expansions and compounding may be 20 per cent to 30 per cent. For triple-expansion engines the most economical steam pressure is of course higher than for compound and may be used to advantage up to 180 pounds or more. The gain from using a triple over a compound may be about 5 to 10 per cent or more. These figures are good only for engines under full load and proper point of cut-off. A compound will usually suffer more loss of economy under light load than a simple, and the triple will suffer more than a compound.

Cut-off and Expansion. The best point for cut-off for a simple engine, whether jacketed or not, is about $\frac{1}{3}$ stroke if the engine is noncondensing and about $\frac{1}{6}$ stroke if condensing. The total expansions of a triple engine or a compound is commonly known as

$$\frac{\text{ratio of low pressure to high pressure}}{\text{fractional part of stroke at cut-off in high}}$$

For instance, if the cylinders are as 1:3:8, and cut-off is $\frac{2}{3}$ stroke in high, then the total expansions as conventionally used is $\frac{8}{\frac{2}{3}} = 20$. For the best service of triple engines on land 26

expansions are used. The conditions of service make it impossible to use as many expansions in marine work, hence the relatively poor economy of marine engines. For compound engines 15 expansions seems to be the best in general. Of course it must be understood that the type of engine and conditions of service may necessitate different arrangements, but when such is the case a less economy of steam consumption usually results.

Variation of Load. An engine should be designed to give a fair economy over a reasonable variation of load. Ordinarily if it gives its best economy at normal load there will be a sufficient range. Usually, it happens that the best mechanical efficiency is obtained with a little longer cut-off than that which shows the most economical steam consumption per indicated horse-power. This is because, as the cut-off lengthens, the power increases faster than the frictional losses. It will be evident from this that increasing the cut-off slightly reduces the thermal efficiency, but the gain in mechanical efficiency may offset the thermal loss. Shortening the cut-off causes loss of both thermal and mechanical efficiency. It would then seem that if the engine were given a little longer cut-off than that which would produce the best thermal efficiency at normal load, the power could be reduced with less actual loss. For a slight reduction we should lose mechanical efficiency and gain thermal, and for a slight increase we should gain mechanical and lose thermal. Thus there would be a wider range with good results. There is always more loss by decreasing cut-off below the point of maximum efficiency than by lengthening it. That is, the engine works at a greater disadvantage when running under a light load than when running under a heavy one.

The allowable range of load for a simple engine is greater than for a compound or triple. If the power of a compound is reduced by shortening the cut-off of the high-pressure cylinder without shortening the low proportionally, there is likely to be an uneven distribution of work and, consequently, a wide fluctuation of temperature, which will cause so much condensation as to offset the advantages of compounding. Moreover, the large expansion in the high-pressure cylinder may reduce the admission pressure to the low cylinder so much that the expansion in the low may be carried below the atmosphere in a noncondensing engine and thus

cause a loop at the end of the card, as we have learned in "Steam Engine Indicators." This loop means a loss of power, and that the high-pressure piston is dragging the low and the engine is using up power on itself as it were. A triple engine is even more troublesome than a compound, and besides giving the trouble already mentioned, the dragging of the low-pressure piston may injure the cylinder and loosen the guides. There is little difficulty in increasing the power of a compound or triple. A simple engine may easily be run at reduced power, either by shortening the cut-off or reducing the steam pressure or both.

Effect of Speed. The transfer of heat between the steam and walls of the cylinder, although very rapid, is not instantaneous. The longer the steam can remain in contact with the cylinder walls, the more heat will be lost. Hence it is reasonable to suppose that an increase of speed will reduce condensation. This has been found by tests actually to be the case. Other things being equal, a reasonable increase of speed results in better economy, but as we can seldom get as good a valve motion at high speed, we may lose almost as much in this way as we gain by decreasing our condensation. The most efficient valve gears are to be found on slow-speed engines, and hence we usually find that the most economical engines are slow speed. Nevertheless, with a given valve gear, an increase of speed gives better economy up to the limit of good valve motion. High-speed engines also require more clearance, which means another loss. These two factors, valve motion and clearance, limit our increase of speed beyond a certain point, just as condensation will limit our indefinite increase of boiler pressure.

Feed Water Heaters. The use of feed water heaters has been discussed in the instruction papers on boilers, but it may be well to observe here that in many places where water is expensive condensing engines cannot be run; nevertheless, a very considerable saving can be effected by allowing the exhaust steam to condense in a feed water heater, and thus save the heat that would otherwise be wasted, or the exhaust steam may be used for heating purposes. Of course in such cases the steam consumption of the engine is high, but if proper allowance is made for the heat used for other purposes, the actual fuel consumption rightfully charged to the engine is not excessive. If the feed water is heated by

waste gases, then the gain belongs to the boiler and not to the engine.

The following represents the average steam consumption in pounds per I. H. P. per hour for various types of engine:

Triple expansion stationary	11 to 14 pounds
Triple expansion marine	13 to 16 pounds
Compound stationary	12 to 15 pounds
Compound marine	18 to 21 pounds
Simple condensing	17 to 20 with jacket
Simple condensing	18½ to 22 without jacket
Simple noncondensing	24 to 33 or more

All these figures have been taken from the results of careful tests of engines in actual service, and represent good engines of their respective types.

Direct-acting steam pumps, although very important engines, are extravagant in their consumption of steam. The absence of a fly wheel makes it almost impossible to use steam expansively, and hence the pump takes steam full stroke.

The steam consumption for a simple direct-acting steam pump may be anywhere from 60 to 200 pounds per horse-power per hour, depending upon the size and conditions of service.

The economy of an engine is usually stated in terms of steam per horse-power per hour, or of B. T. U. per horse-power per minute. The latter is more useful for purposes of comparison with other engine tests, but the real criterion of economy is the actual cost of power in terms of fuel consumption, for this can at once be reduced to dollars and cents, and the manufacturer or the builder of the engine can determine the effect of devices for economy in terms of money, which to him is worth more than B. T. U. It may often happen that some expensive device, while effecting a saving of heat consumption, may not save fuel enough to pay for its first cost, owing to the peculiar conditions of service.

TESTING.

The principal object in testing a steam engine is to determine the cost of power or the effect of such conditions as superheating, jacketing, etc., upon the economy of the engine. We must therefore, in the first case, measure the cost of fuel, and in

the second the actual heat used. In either case we must calculate the power of the engine.

The indicated power is determined in the familiar way by means of the indicator, and the actual power by means of a dynamometer or friction brake. (For further particulars of these instruments and apparatus see "Steam Engine Indicators.") To determine the cost of power in terms of coal it is necessary to conduct a careful boiler test, usually of twenty-four hours duration.

When the cost is expressed in terms of steam per horsepower per hour, we may follow either of two methods. We may condense and weigh the exhaust steam, or we may weigh the feed water supplied to the boiler. An hour under favorable conditions is usually sufficient for such tests. Steam used for any purpose other than running the engine must be determined separately and allowed for.

Probably the most accurate terms in which to state the performance of an engine is in B. T. U. per horse-power per minute. When the cost is expressed thus, it is necessary to measure the steam pressure, amount of moisture in the steam, and temperature of condensed steam when it leaves the condenser. Jacket steam must be accounted for separately. Important engines with their boilers, etc., are usually built under contract to give a certain efficiency, and their fulfilment of this contract can be determined only by a complete test of the entire plant. Before beginning the test, the engine should be run for a short time in order to limber it up and get it thoroughly warmed. It is of the utmost importance that all conditions of the test should remain constant, especially the boiler pressure and the load. All instruments used in the test should be tested themselves before being used, in order to determine the effect of any errors to which they may be subject.

Thermometers. All important temperatures, such as feed water, injection water, condensed steam, etc., must be taken by reliable thermometers, the errors of which have been previously determined and allowed for. Good thermometers sold by reliable dealers are usually satisfactory. Thermometers with detachable scales are subject to serious errors, and should be used only for very crude work. Cheap thermometers are of little value in an engine test.

Indicators. The most important and least satisfactory instrument used in the test is the indicator. It is subject to an error of 2 per cent or 3 per cent at low speed, and this may easily be two or three times as much at high speed. It does not work satisfactorily at more than 400 revolutions per minute. If the indicator is carefully tested under conditions similar to those on the engine, the errors may be reduced to a minimum, but there will always be some uncertainty. The principal errors to which the indicator is subject have been mentioned in the instruction paper on "Steam Engine Indicators." It may, however, be well to add that for accurate work we should always use two indicators, as the long piping and joints necessary for only one causes a considerable loss of pressure and much condensation. For marine work it is customary to use only one indicator, with a three-way cock, the lower end of the cylinder usually being inaccessible.

Scales. Weighing should be done on standard platform scales. The water may be weighed in barrels provided with large drain valves which will allow the water to run out quickly. It is seldom possible to drain barrels completely, and so it is best to let out what will run freely, then shut the valve and weigh the barrel. This we call "empty" weight, and deducted from the weight "full," evidently gives us the true weight of water.

If not convenient to weigh the water, it may be measured in tanks or receptacles of known capacity, and the temperature taken, allowing the proper weight per cubic foot for water at that temperature; or it may be determined by meters.

Meters. Water meters are of two kinds, those that record the amount of water by displacement of a piston, and those in which the flow is recorded by means of a rotating disc. Piston water meters can be made very accurate, and if working under fair conditions of service they may be relied upon to a close degree. The chief error in a meter arises from the air that may be in the water. To reduce this error to a minimum, the meter should be vented, to allow the air to escape without passing through the meter. Rotary meters are good enough for very rough work, but are seldom sufficiently accurate for a careful engine test. So far as possible, weirs should not be used in engine work. They may be fairly accurate under certain conditions,

[illegible]

but a very little oil in the water may affect them seriously. They may sometimes be used to measure the discharge from a jet condenser, for then the volume is so large that the actual error is proportionately small.

Gages. Pressures should be measured on good gages that have been recently tested by comparison with a mercury column. The atmospheric pressure should be read from the barometer, and for accurate work this pressure should be used. For ordinary work, 30 inches, or 14.7 pounds, will do.

Calorimeter. When using superheated steam it is sufficient to take the temperature and pressure in the steam pipe, but if saturated steam is used, we must determine the amount of moisture it contains. This is done by means of a calorimeter such as described in "Boiler Accessories."

Brake. Any of the forms of friction brake described in "Steam Engine Indicators" will answer the purpose. For smooth and continuous running it is essential that the brake and its band be cooled by a continuous stream of water. The water may either circulate in the rim of the wheel or around the brake band, but it must not come in contact with the rubbing surfaces.

If the load is steady, seven or eight observations at equal intervals will usually be sufficient. If possible, the cards should be taken simultaneously, and then all the data averaged for the final result. If the load fluctuates, the cards must be taken oftener, and a greater number of observations will be required. The greatest care and accuracy must be used in all this work. In conducting a test, a careful log should be kept of the data; the outline given on page 56 being a suggestion.

THE STEAM TURBINE.

A description of the steam turbine and the general principles of the engine were given in Part I of "The Steam Engine." Now we shall discuss its efficiency. The turbine is such a comparatively new engine that there have been but few tests made, and consequently we know very much less about its possibilities than is the case with the ordinary reciprocating engine.

Probably the greatest loss in the reciprocating engine is that due to condensation and the subsequent re-evaporation at exhaust.

The exhaust cools the cylinder, so that the incoming steam meets cool walls, while in the turbine no such conditions exist. The admission takes place at one end of the engine and the exhaust occurs at the other end. The temperature gradually falls from admission to exhaust and the expanded steam never comes in contact with the part in which the higher-pressure steam works. Thus Watt's principle seems to be fulfilled, namely, "The cylinder should always be as hot as the steam that enters it." Of course there is considerable loss from radiation, but there should be very much less condensation than in the reciprocating type.

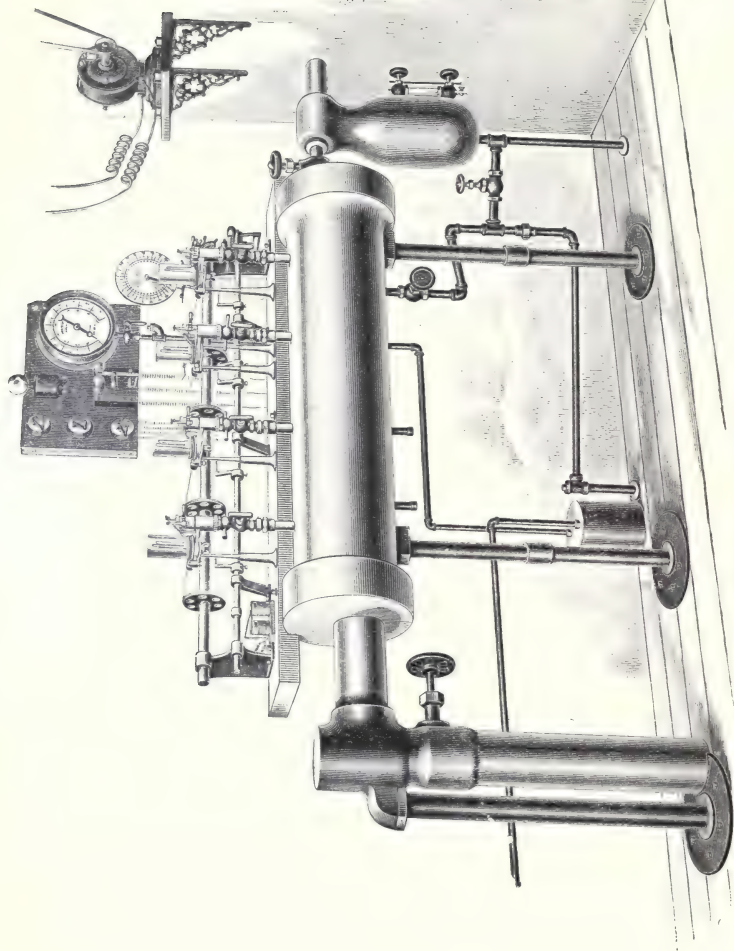
Superheating may be made use of with considerable gain in economy. There are no rubbing surfaces, no lubricants to decompose and no glands to burn out, as is the case in the reciprocating engine. The steam may be superheated to 60° or more to advantage. There being no internal lubrication, there is of course no oil to get into the condenser, and so the feed water may be used without fear of getting grease into the boiler.

Another advantage seems to be in the more complete expansion of the steam. There is little gain in the reciprocating engine by expanding the steam beyond a certain limit, because of the increased condensation. The boiler pressure cannot be increased indefinitely, neither can the expansion be carried out to the limit.

From these considerations it would seem as if the turbine ought to show much better efficiency than the reciprocating engine, and were it not for the friction of steam against the vanes of the turbine the advantage would doubtless be in its favor. Tests have shown a consumption of about 16 pounds of steam per B. H. P. per hour. Assuming an efficiency of 85 per cent, this would give about 14 pounds per I. H. P. per hour. Tests of the best modern triple expansion pumping engines have shown a steam consumption of a little over 11 pounds per I. H. P., and numerous tests of ordinary triple expansion engines have been made which show a consumption of 12 to 13 pounds.

The most recent test of which we have accurate knowledge was made on a Westinghouse-Parsons engine of 600 H. P. when running at 3,600 revolutions per minute. With ordinary steam having 3 per cent of priming, the turbine used 15.5 pounds of steam per B. H. P. per hour with a vacuum of $25\frac{7}{8}$ inches in the

condenser. With 30° of superheating and $26\frac{1}{2}$ inches of vacuum, the steam consumption fell to 14.2 pounds. Assuming 85 per cent mechanical efficiency as before would give a relative steam consumption of 13.17 pounds per I. H.P. in the first case, and 12.07 pounds in the second case. It will be instructive to compare these figures with those given for reciprocating engines on page 53.



MERCURY COLUMN FOR CALIBRATING INDICATOR SPRINGS.
American Steam Gauge and Valve Manufacturing Company.

STEAM ENGINE INDICATORS

A most important question concerning a steam engine is, "What is its horse-power?" or "How much work will it do in a given time?"

Work is defined as *pressure, force, or resistance* multiplied by the *distance* through which it acts.

Power is work done in a specified time.

In the steam engine, steam is the agent by means of which heat is transformed into mechanical work. It is the heat in the steam that does the work, not the steam itself.

Work is obtained from the heat in steam by confining it in a closed cylinder which is fitted with a piston and a piston-rod. Steam is admitted at one side of the piston while the other is open to the atmosphere or in communication with a condenser. The pressure of steam, usually 75 to 150 pounds per square inch, forces the piston to the other end of the cylinder, driving out the low-pressure steam in front of it. When it arrives at the other end, steam is admitted to that end and the piston is driven back.

The piston moves because the pressure on one side is greater than that on the other.

In order to move the piston, work must be performed. The amount of work is easily found, since work equals total pressure multiplied by the distance through which the piston moves.

Suppose a piston is 2 square feet in area and steam at a pressure of 64.7 pounds per square inch acts on it during the entire stroke of 4 feet; the other side of the piston being in communication with the atmosphere. The total pressure is then $2 \times 144 \times 64.7 = 18,633.6$ pounds. If this pressure acts through 4 feet it is evident from the definition that the work done per stroke will be,

$$18,633.6 \times 4 = 74,534.4 \text{ foot-pounds.}$$

Another method is as follows: The pressure on the above piston is $64.7 \times 144 = 9316.8$ pounds per square foot. The volume swept by the piston during one stroke is $2 \times 4 = 8$ cubic feet. If we multiply the pressure per square foot by the volume or 9316.8 by 8, we get 74534.4 foot pounds, the same result as before. Thus we see that work equals **unit pressure** multiplied by **volume**.

Let P = pressure on the piston in pounds per square foot.

p = pressure on the piston in pounds per square inch.

A = area of piston in square inches.

L = length of stroke in feet.

V = volume swept by piston in one stroke in cubic feet.

W = work done in foot-pounds.

Then from the above example,

Work = unit pressure multiplied by volume.

or, $W = P \times V$

It is evident that $P = 144 p$, and $V = \frac{A}{144} \times L$.

Then we have these expressions for work,

$$W = P \times V = 144 p \times V = 144 p \times \frac{A}{144} \times L = p L A.$$

Suppose steam is admitted to the cylinder during the whole stroke, as in the above example, that is, one end of the cylinder is in communication with the boiler. The other end is open to the atmosphere. If we draw two lines at right angles to each other, as OY and OX in Fig. 1, the volume of steam for any position may be represented by some distance measured on the line OX . Similarly the pressure of the steam at any position of the piston may be represented by the length of a vertical line parallel to the line OY .

In the above example, the area of the piston was 2 square feet, the length of stroke 4 feet and the pressure by gage 50 pounds. Then we let OA = the atmospheric pressure = 14.7 pounds. At the beginning of the stroke the pressure (absolute) is $14.7 + 50 = 64.7$ pounds, represented by the distance OB , or $AB = 50$ pounds pressure. When the piston has passed through $\frac{1}{4}$ of the stroke it is represented as the point 1, or $B1$ is the volume swept through when the piston has completed $\frac{1}{4}$ of the

stroke. At this point the pressure is also 64.7 pounds as represented at 1'. Similarly, when the piston is at 2, 3, and 4 the corresponding pressure is 2', 3', 4'. Since the pressure is constant the line B D is parallel to O X. We see from the above that 50 pounds is the net pressure acting on the piston during the stroke, and is represented by A B and lines parallel to it. The volumes are represented by the horizontal line A C. Then since $W = P \times V$ it also equals O B $\times$ O X which is evidently the area of the rectangle O B D X. The area of the rectangle O B D X is proportional to the work done by the steam.

In Fig. 1, one inch on the line O Y = 40 pounds, then O B is 1.6175 inches long since it represents 64.7 pounds. Similarly O A must be .3675 inch since it represents 14.7 pounds. The line A C is 2 inches long; then referring to the preceding example, one inch in length = $\frac{8}{2} = 4$ cubic feet.

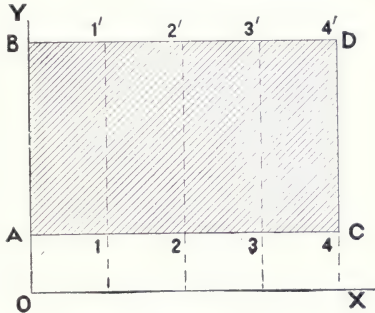


Fig. 1.

Since the rectangle O B D X is 1.6175 by 2 inches, the area is 3.235 square inches. But one inch in height equals 40 pounds pressure and one inch in length equals 2 cubic feet. Then $p V = 40 \times 3.235 \times 4 = 517.6$ foot-pounds and, $W = 144 p V = 517.6 \times 144 = 74,534.4$ foot-pounds.

In the above cylinder the pressure acting on one side of the piston was 64.7 pounds per square inch. There was also a pressure of 14.7 pounds per square inch (the atmospheric pressure) acting in the opposite direction. Then the work done against the steam pressure is represented by the area O A C X and is equal to $144 p V = 144 \times 14.7 \times 8 = 16934.4$ foot-pounds. Then since O B D X represents the total work done on one side of the piston and O A C X represents the work done *against* the piston the difference A B D C represents the **net work**. This net work is represented by the shaded area. Also if the amount of work done

on the piston is 74,534.4 foot-pounds and the work done against the piston is 16,934.4 foot-pounds, the net work is the difference, or 57,600 foot-pounds.

In this theoretical discussion the same result may be obtained by subtracting the atmospheric pressure or back pressure from the absolute initial pressure and using the difference as the value of p . This value of p is called the mean effective pressure.

Then $64.7 - 14.7 = 50$ and

$W = 144 p V = 144 \times 50 \times 8 = 57,600$ foot-pounds.

The area is proportional to the work done whatever the shape may be; provided the line B D represents the relation between pressures and volumes on the steam side of the piston and the lower line A C represents the relation between pressures and volumes on the exhaust side. If the engine is of the condensing type the line A C will be nearer O X, which is the line representing *absolute vacuum*.

Whatever the shape of the diagram, the area is equal to the area of a rectangle of the same length and a height equal to the mean height, or mean ordinate as it is called. The mean ordinate represents the mean or average net pressure on the steam side of the piston. Then we can follow these rules in finding the work of the steam from the diagram.

Multiply the area in square inches by the scale of pressures, by the scale of volumes and by 144, or ;

Multiply the length of the mean ordinate by the scale of pressures, by the length of stroke, and this product by the area of the piston in square inches.

Example: The area of a diagram A B D C like that of Fig. 1 is 6.3 square inches and its length is 3 inches. The scale of pressure is 30 pounds per inch and the scale of volumes is 1.99985 cubic feet to the inch. If the piston is 20 inches in diameter and the length of stroke $2\frac{3}{4}$ feet, what is the work done per stroke?

Solution:

$W = \text{area of diagram} \times \text{scale of pressures} \times \text{scale of volumes} \times 144.$
 $= 6.3 \times 30 \times 1.99985 \times 144 = 54,428$ foot-pounds.

$W = \text{mean ordinate} \times \text{scale of pressures} \times \text{area of piston} \times$
 length of stroke.

$= 2.1 \times 30 \times 314.159 \times 2\frac{3}{4} = 54,428$ foot-pounds.

Thus we see that we get the same result by both rules. The latter is the more common method because the mean ordinate is easily found and the scale of volumes seldom considered.

In our consideration of Fig. 1, steam was admitted to the cylinder during the *entire stroke*. In modern engines this method is rarely used; instead, steam is admitted during *part* of the stroke then the communication to the boiler is cut off, and the steam in the cylinder allowed to **expand**, as the piston moves forward, until it fills the entire volume of the cylinder. This is represented graphically in Fig. 2.

Steam is admitted to the cylinder until the piston reaches the point 2 which represents one-half the volume of the cylinder. Then the cylinder is half full of steam, that is, it contains $\frac{8}{2} = 4$ cubic feet. The four cubic feet of steam expand until they fill the cylinder. Since there is the same weight of steam present at every point in the stroke and the volume continued to increase, the pressure must diminish.

This is shown in Fig. 2. The line B 2' is horizontal because the pressure remains constant to the point of cut off. Then the pressure begins to fall as is represented by the curved line 2' E. This curve is nearly an equilateral hyperbola.

From Fig. 1 we know that the area B 2' 2 A is proportional to the work done while the piston moves from A to 2 or during the first half of the stroke. If we use the same data as we did in Fig. 1, the work done must be one-half the work done in the first case, or $\frac{57600}{2} = 28800$ foot-pounds. Also the area B 2' 2 A is easily found since it is a rectangle. The area 2' E C 2 is found by dividing it up into small sections, by calculus or by the use of a planimeter.

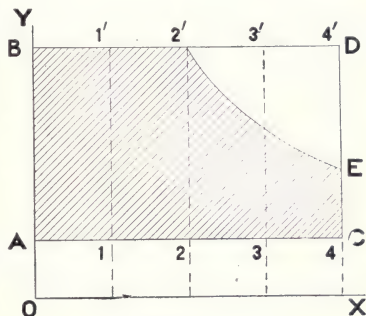


Fig. 2.

It is easily seen that the area of the second case Fig. 2, is less than that of Fig. 1. Therefore the work done is less; but the amount of steam admitted is only one-half as much as in the first case.

In the first case, Fig. 1, 8 cubic feet of steam at 50 pounds pressure were admitted per stroke and the work done was found to be 57600 foot-pounds. In the second case only half as much steam is admitted and the work done is $\frac{57600}{2}$ + the amount represented by the area 2' E C 2. Thus we see that there is a considerable gain by expanding the steam.

Watt's Diagram of Work. Fig. 3 illustrates the method adopted by James Watt to show the action of steam in the cylinder

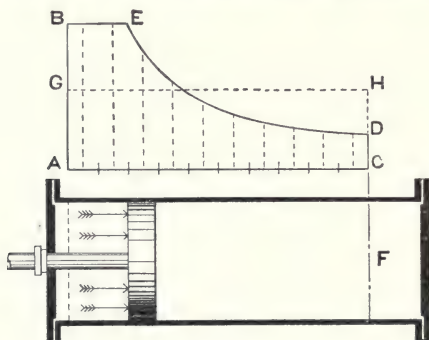


Fig. 8.

der. The horizontal line A C called the abscissa represents the length of the stroke and is divided into ten equal parts. The vertical line A B called the ordinate indicates the pressure of steam.

When the piston has moved to the point E steam is cut off, that is, a volume of steam equal to $\frac{1}{5}$ the volume of the cylinder expands until it fills the entire cylinder. The area may be found by adding the several pressures (shown by the dotted lines), dividing by the number of divisions, and multiplying by the length.

If by some arrangement of steam tight pistons working in cylinders and having pencils fastened to them, we could get a dia-

gram like that shown in Fig. 3 it would be of great use but too large for convenience.

To obtain the same diagram on a small scale an indicator is used. The value of such a diagram has already been shown when finding the work done in the cylinder. The indicator has enabled engineers to bring the engine of today to its present state of excellence. A correct idea of the action of steam in the cylinder can be obtained only by means of an indicator. It shows whether or not the valves are set properly and how the condenser is working. It also shows the engineer which end of the cylinder is doing the most work. By comparing the expansion line with an equilateral hyperbola, with a curve of constant steam weight, or with an adiabatic curve for steam, the cylinder condensation is calculated.

James Watt was the first to see the need of accurate knowledge of the action of steam in the cylinder. He invented the indicator. The improved form consisted of a steam cylinder S, about one inch in diameter and six inches long, in which a solid piston P, is accurately fitted. A spiral spring A, is attached to this piston, and controls the motion of a pencil *a*, which is also attached to the piston. This pencil can operate on a sheet of paper fastened to a sliding board, B. This board moves back and forth by means of a weight at one end and a cord at the other which is connected to some reciprocating part of the engine. The indicator cylinder S, may be put in communication with the engine cylinder by means of the cock C. With this instrument a complete diagram can be taken.

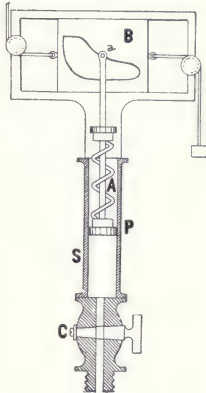


Fig. 4.

Watt's first indicator had no lateral motion, therefore all it showed was the pressure of steam in the cylinder and the perfection of the vacuum.

INDICATORS.

The diagram, or card as it is often called, obtained by the use of an indicator is the result of two motions. The *horizontal move-*

ment of the paper corresponds exactly to the movement of the piston, and the *vertical* movement of the pencil is an exact ratio to that of the pressure of steam in the cylinder. The diagram represents by its *length* the stroke of the engine and by its *height* the steam pressure on the piston at the corresponding point of the stroke. The diagram shows the action of steam on one side of the piston only; to obtain the same information in regard to the other side it is necessary to take another diagram from the other end of the cylinder.

The essential features of an indicator are found in the instrument invented by James Watt. Since his time, however, the many improvements have made the indicator light, compact, durable, and accurate. Watt's diagram was traced on paper stretched on a sliding board but now a revolving drum is used. The height also of Watt's diagram was equal to the movement of the spring, and the pencil arrangement was a simple contrivance. In the indicators of the present day, the spring has a slight movement, the height of the card being obtained by a multiplying arrangement of levers. This method requires a parallel motion to obtain accuracy in the vertical lines; for if a lever is pivoted at one end and power applied near the pivot the lever tends to rise and the free end will describe an arc of a circle, not a straight vertical line.

THE THOMPSON INDICATOR.

Two views of the American Thompson Indicator, the outside and the inside, are shown in Figs. 5 and 6. The form of spring is shown in Fig. 7. The indicator consists of a cylinder in which a piston is fitted, a spring, multiplying lever and parallel motion for the pencil and a cylinder or drum for the paper. The piston, which is .798 inch in diameter = $\frac{1}{2}$ square inch in area, is fitted accurately to the cylinder and has a travel of about one-half inch. When the pressure of steam forces the piston upward it compresses the spring above it; the amount of compression varies with the strength of the spring. The rise of the piston causes the pencil to rise because of the system of levers. The cylinder to which the paper is attached rotates by means of a cord which is fastened to some part of the engine, the crosshead for example.

While the drum revolves and the steam pressure forces the piston to rise, the pencil, lightly touching the paper, describes the diagram.

The parallel movement of this indicator is obtained by a link attached directly to the lever. It is so constructed that there is but little lost motion, hardly any friction and no appreciable error within the limited movement of the pencil.

The paper cylinder is so constructed that the tension of the coiled spring within the drum may be altered for different speeds of the engine. By this means the cord can be kept taut with little trouble. The cord is led through a hold and kept in contact with the scored wheel by another small one. By this means the cord can be run to any angle. It is convenient to have the cards of about the same size. If we used a

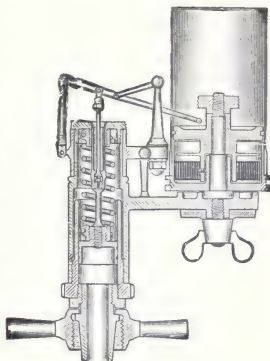


Fig. 5.

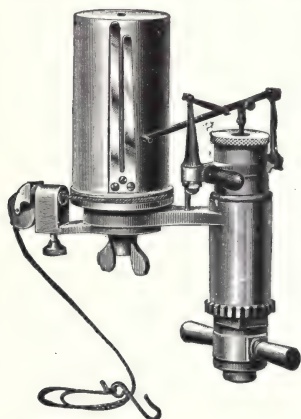


Fig. 6.

To Change the Springs. In selecting the spring for a given pressure, care should be taken that it will easily stand that press-

spring of such a tension that the pencil would move one inch for every 60 pounds pressure, and there was but 30 pounds pressure in the engine cylinder, the diagram would be but one-half inch high. This diagram would be too small for accurate work. For this reason indicators are provided with sets of springs of varying stiffness which may be used according to the steam pressure. If a spring is of such stiffness that the pencil moves 1 inch for every 50 pounds pressure it is called a 40 pound spring. Others are called 10 pound, 20 pound, 30 pound, etc., springs.

ure. A safe rule to follow: multiply the scale of the spring by $2\frac{1}{2}$ and subtract 15 for the vacuum.

For example: The maximum pressure for a 50 pound spring is 110 pounds, because $(50 \times 2\frac{1}{2}) - 15 = 125 - 15 = 110$. Springs are made in the following scale: 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 56, 60, 64, 80, 100. For pressures from 70 to 90 pounds a 40* pound spring should be used, as 80 pounds pressure on a 40 pound spring will raise the pencil 2 inches, and this is a good height for the diagram.

If very high pressures are to be indicated, an extra piston, having an area of $\frac{1}{4}$ square inch, is used. This doubles the allowable pressure on the spring. For instance, if a spring can be used for 110 pounds pressure when the piston is $\frac{1}{2}$ inch in area, it can be used for 220 pounds pressure if the $\frac{1}{4}$ inch piston is used.

When the spring has been selected it is placed in the indicator. First unscrew the milled nut at the top of the steam cylinder and take out the piston with arm and connections. The pencil lever and piston are disconnected by unscrewing the small-headed screw which connects them. If a spring is connected to the piston it should be removed, the selected one substituted and the indicator put together. The spring should always be firmly screwed to the shoulder or the pencil will not properly indicate the pressure.



Fig. 7.

Care of the Indicator. Before attaching the indicator to the engine cylinder it should be taken apart, cleaned and oiled. If each part works freely and smoothly the spring may be put in and the indicator put together. After connecting to the cylinder, admit steam to it, but do not take cards until it is thoroughly warmed and blows dry steam through the relief. It is not necessary to use lead in connecting as it is likely to get into the indicator. After using, take the indicator apart, clean and oil. Only porpoise or fine watch oil should be used.

THE CROSBY INDICATOR.

The internal arrangement of the parts of the Crosby Indicator

•NOTE In practice it is better to use a somewhat stiffer spring.

is shown in Fig. 8. The piston, 8, is formed with shallow channels on its outer surface to retain oil which prevents leakage and lubricates the piston.

The socket in the center of the piston is supported by a central web and projects both upward and downward. The upper

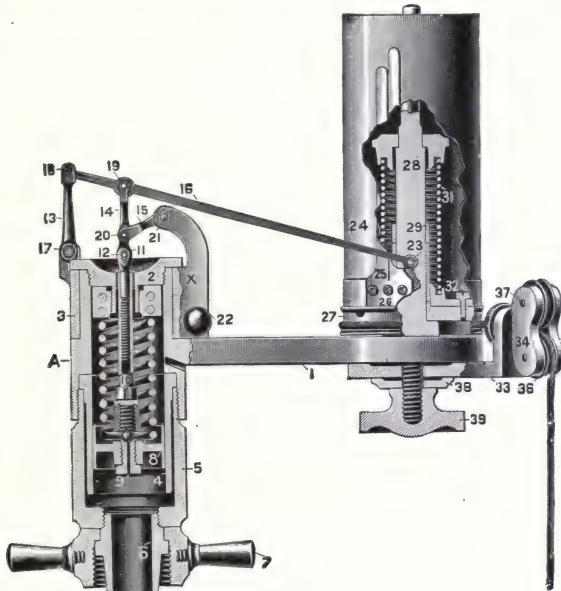


Fig. 8.

portion is threaded inside to receive the lower end of the piston-rod. It has a vertical slot which allows the ball bearing on the end of the spring to drop into a concave bearing on the upper end of the piston-screw 9 which is screwed into the lower part of the socket.

The piston-rod, 10, is made hollow, with the lower end threaded. When the piston-rod is connected to the socket, the former should be screwed into the socket as far as it will go.

The height of the atmospheric line on the diagram depends

upon the amount the swivel head, 11, is screwed into the top of the piston-rod.

A small projection on the lower side of this cap is threaded to screw into the top of the spring and hold it firmly in place. The moving parts are kept in line by this cap. The pencil mechanism is supported by the sleeve 3, which surrounds the upper part of the cylinder; it turns freely and is held in position by the cap.

The pencil mechanism is made as light as is consistent with strength and stiffness. The pencil moves exactly parallel to the piston because the fulcrum of the mechanism and the point of attachment to the piston-rod are always in a straight line. The pencil point moves six times as far as the piston because of the multiplying levers.

The drum 24, is one and one-half inches in diameter, and is rewound when the string is pulled, by a short spiral spring 31.

The piston spring is made of a single piece of steel wire wound from the middle into a double coil. The ends are screwed into a brass head having four radial wings. At the bottom of the spring a small steel bead is firmly attached to the wire. This forms a ball and socket joint with the lower end of the piston-rod. This joint is light and allows the spring to yield to pressure from any direction. These springs are made in the following scale: 8, 12, 16, 20, 24, 30, 40, 50, 60, 80, 100, 120, 150, and 180.



Fig. 9.

To Insert the Spring. First unscrew the cap 2, then lift the connected parts free from the cylinder by means of the sleeve. The hollow wrench should be held in an inverted position and the piston-rod inserted until the hexagonal part engages the wrench. Then, having the spring shown in Fig. 9 inverted, insert the combined wrench and piston-rod until the steel bead and the end of the spring rests in the concave seat. Now invert the piston and pass the transverse wire at the bottom of the spring through the slot until the threads at the bottom of the piston-rod engage those inside the socket of the piston. With the wrench screw it in as far as it will go.

The piston screw should be loosened slightly before the piston-rod is screwed in, and afterward set up against the head lightly to prevent lost motion. Then with the sleeve and cap upright, engage the threads of the swivel head with those inside the piston-rod and screw it up until the lower projection of the cap engages the threads inside the spring top; continue the process until the spring is screwed up firmly against the cap. Holding only by the sleeve 3, turn the piston and the connections until the top of the piston-rod is flush with the shoulder on the swivel head.

Now that the piston and all the connections are in their places, the whole may be inserted in the cylinder and the cap screwed down, which will fix all parts in their proper places. If there is a spring in the cylinder first detach by reversing the above process.

THE TABOR INDICATOR.

The Tabor Indicator is shown in Fig. 10. It is used extensively in the navy. The principle of action and details of construction are similar to the indicators already described; the chief peculiarity being the means employed to obtain a straight line movement for the pencil. Inside the steam cylinder is a lining in which the piston moves. This lining can expand when heated. In the side of the cylinder small holes allow any steam which may leak by the piston to escape. The piston-rod is connected at one end to the piston by means of a ball and socket joint; the other end is connected to the pencil mechanism.

The **pencil mechanism** consists of three pieces, the pencil lever, the back link, and the piston-rod link. The two links are parallel for every position of the pencil. Thus the lower pivots of these links and the pencil point are always in the same straight line. The straight line movement of the pencil is obtained by means of a curved slot in a stationary plate. The pencil bar is provided with a roller which is fitted in such a manner that it can roll from one end of the slot to the other. The curve of the slot guides the pencil bar and is of such a radius that the pencil is caused to move in a straight line. The curve compensates for the tendency of the pencil point to move in an arc of a circle. The

pressure of the pencil on the paper is regulated by a screw which strikes a stop plate attached to the frame. The end of the pencil bar is formed for either a pencil lead or a metallic marking point.

The drum for the paper is made similar to those of other indicators. The backward movement is obtained by a flat spiral spring placed under the drum. The tension of this spring is altered by loosening a thumbscrew, lifting the carriage, and winding or unwinding. A simple pulley guides the driving cord in any direction.

The indicator is attached by a coupling having a single thread.

The **springs** of the Tabor indicator are of the duplex type, that is, they are made of two spiral coils of wire. A 50 pound spring is shown in Fig. 11. The wire terminates in fittings at each end. The spring is attached to the upper side of the piston by means of threads cut on the inside of the fitting and on a projection on the piston. The top of the spring is attached to the under side of the cover in a similar manner. The springs are made in the following scales, 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 60, 64, 80, and 100 pounds. The maximum safe steam pressures (absolute) to which these springs may be subjected are respectively, 10, 15, 20, 24, 40, 48, 70, 75, 95, 112, 120, 140, 152, 180, and 200 pounds.

Change of Location of Atmospheric Line. Unscrew the cap and lift the sleeve and connections from the cylinder. Then turn the piston to the left or right according as the pencil is to be raised or lowered. One revolution causes the pencil to rise $\frac{1}{2}$ inch.

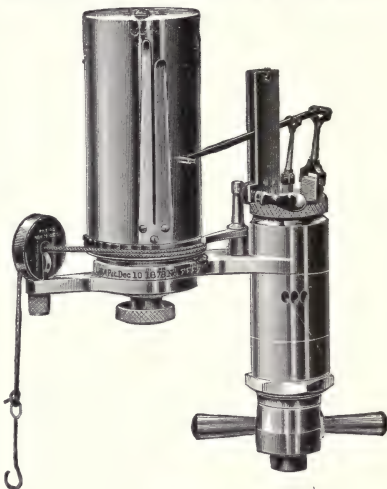


Fig. 10.

Care of the Indicator. Before attaching the indicator to the engine, steam should be blown through the pipes and cocks so that all particles of dust may be removed. After using, the indicator should be carefully wiped and oiled. The cylinder cap should be unscrewed and all the parts connected to the piston removed. The piston spring and piston-rod should be detached, carefully wiped dry, and then oiled. The inside of the cylinder also should be oiled. Then the piston and piston-rod should be placed in the cylinder and the spring placed in the box. If the indicator has not been used for some time the oil may have become



Fig. 11.

gummed. It may be easily cleaned by wiping with a cloth saturated with naphtha or benzine. It must be oiled again before using. A good test that the indicator is in proper working order is to detach the spring and after replacing the piston and piston-rod, raise the pencil to the highest point. When allowed to fall it should descend to the lowest point freely. The pencil should always have a smooth fine point.

To Attach the Indicator to the Engine. Usually all first-class engines are prepared for the indicator before leaving the factory. Holes are drilled and tapped in the cylinder and have plugs screwed in them. These plugs are easily removed and the indicator connections screwed in.

When this is not the case, any engineer can perform the work. Before drilling the holes, in the cylinder, the heads should be removed so that the exact positions of the pistons and the size of the ports and passages may be known. Also with the heads off all chips and particles of dirt from drilling may be easily removed. If it is impossible to remove the heads, a little steam admitted to the cylinder just before the drilling is completed will blow the chips out.

Each end should be drilled and tapped for a one-half inch pipe thread. The holes must be drilled into the clearance space, so that the piston at the ends of the stroke will not cover them. They should also be placed so that currents of steam will not reach them. Before deciding just the points at which to drill the holes, it is well to consider every plan of indicating the engine. The type of engine, the position of the steam chest, the kind of cross-

head, and the position of the eccentric and its connections should all be considered, as well as the most convenient place in the engine room. The holes should not be drilled until the plan shows the proper connections with the reducing motion, convenient access and free passage of steam to the indicator.

When the plan has been adopted, the engine should be placed on dead center, to determine the clearance. The holes should be drilled into the middle of the clearance space.

In common practice for horizontal engines the holes are drilled in the side of the cylinder at each end. Short half inch pipes with quarter upward bends into which the indicator coils may be screwed are inserted in these holes.

It may be more convenient to drill and tap into the top of the cylinder and attach the indicators directly.

For vertical engines, the upper head or cover and the side of the cylinder are often drilled and tapped for the upper and lower indicators respectively. It is preferable to connect the indicators

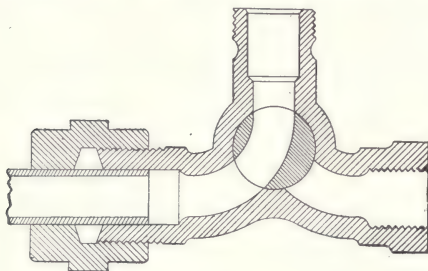
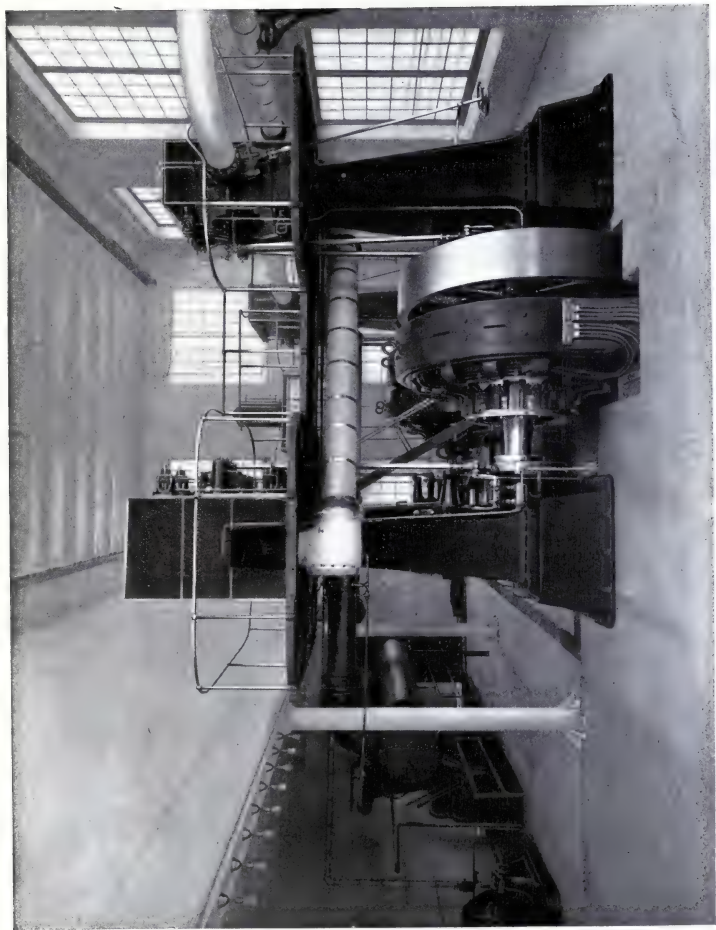


Fig. 12.

to the sides because less pipe and fittings are required and better results obtained.

If only one indicator is to be used for both ends of the cylinder, it may be connected by side pipes and a **three way cock**. By this method both diagrams are taken on the same card and with the loss of but one revolution. Fig. 12 shows the section of a three way cock.

Reducing Motion. As we have already seen the length of the card represents the travel of the piston. As the length of



VERTICAL CROSS-COMPOUND CORLISS ENGINE IN THE NEW YORK AND PENNSYLVANIA RAILWAY PLANT
AT JOHNSONBURG, PENNA.
Ball Engine Co., Erie, Penna.

card is obtained by the rotation of the drum the motion of the drum must be taken from some part of the engine which has a motion coinciding with that of the piston. The crosshead is the most common, reliable and convenient part. The length of the card is much less than the travel of the piston since the stroke is longer than the circumference of the drum so that the movement of the crosshead must be reduced to the length of the diagram.

There are several devices employed to obtain this reduced motion.

A simple form of reducing motion, called the **pantograph**, is shown in Fig. 13. Four links, *a*, *b*, *c*, *d*, are joined in the form of a parallelogram. One link, *a*, is prolonged and pivoted at the crosshead C. The point where *b* and *c* join is pivoted at the fixed point E. The cord is fastened at D on the link *d*.

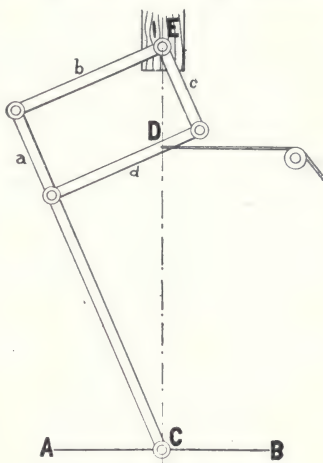


Fig. 13.

The point D must be in the straight line connecting E and C. Then letting *AB* represent the stroke and *h* the length of the indicator diagram, we have

$$AB : h = EC : ED, \text{ from which}$$

$$ED = \frac{h \times EC}{AB}$$

Another form of pantograph is shown in Fig. 14. It is placed horizontally with the pivot, B, resting on a support opposite the crosshead when in mid-position. The pivot A, is attached to the crosshead; usually by having the stud A inserted in a hole drilled in the crosshead. If the pivot B is adjusted to the proper height and at the right distance from the crosshead, the cord from the indicator may be attached to the pin E without any pulleys. The

length of the diagram is varied by adjusting the movable bar C D; the pin E must be in the straight line from A to B.

The pantograph is likely to become shaky and loose on account of its many joints. If well made it gives perfect motion.

The reducing motion shown in Fig. 15, called the Brumby Pulley, is easily and quickly made and can be used on almost any engine. The wooden rod A is usually about twice as long as the stroke. It is pivoted by a bolt or screw at B, a fixed point. At the lower end it is connected by the wooden link, C, to the crosshead.

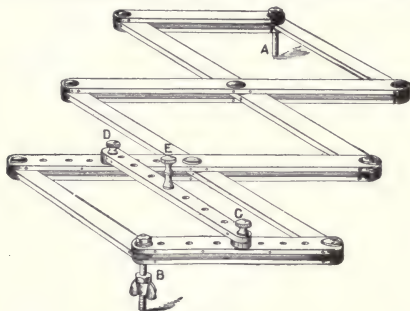


Fig. 14

This link C is usually about one-half the length of the stroke. The sector S may be made either of wood or metal. It should have a groove in the circular edge for the cord, and is made fast to the upper end of the lever A. Its center should coincide with that of the pivot B. The length of the radius of the sector may be found as follows. Divide the length of the lever by the length of the stroke, multiply the result by the length of the desired diagram and the product will be the radius of the sector. For instance, if the lever is 60 inches long and the stroke 30 inches and we wish the diagram to be 3 inches long, the sector should be 6 inches in radius for,

$$\frac{60}{30} \times 3 = 6.$$

To avoid the use of guide pulleys the lever should be hung so that it will swing in a vertical plane parallel with the guides and in line with the indicator. When the crosshead is at mid-stroke,

the lever must be vertical and the point D must be below the axis of the cylinder because it comes above the line at the ends of the stroke.

The reducing lever used for large quick running engines is shown in Fig. 16. The rod A is made of pine wood, tapering toward the lower point and about one inch in thickness. The length is about one and one-half times the length of the stroke. It is suspended by a bolt or screw from some fixed point above the engine, and should swing edgewise and parallel to the guides of the crosshead. The steel stud at the bottom of the rod has a T shaped slot in an iron plate which is attached firmly to the crosshead. The slot should be long enough to retain the stud when the crosshead is at the end of the stroke. To find the point at which the indicator cord should be attached, divide the length of

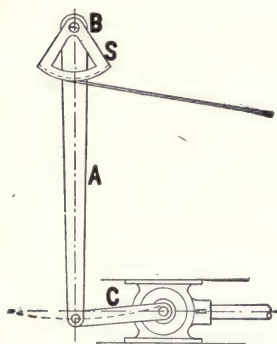


Fig. 15.

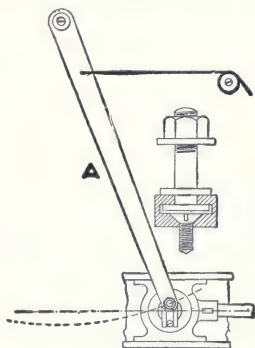


Fig. 16.

the lever by the length of the stroke of the piston and multiply the quotient by the length of the desired diagram. The product is the distance of the point from the pivot at the top of the lever.

Example. A lever is 45 inches long, the piston stroke 30 inches and the diagram to be $3\frac{1}{4}$ inches long. At what distance from the pivot should the indicator cord be attached?

$$\frac{45}{30} \times 3\frac{1}{4} = 4\frac{7}{8} \text{ inches.}$$

Having placed the indicator in position and obtained the

reducing motion, the length of the cord must be so adjusted that the drum will not strike the stops at either end of the stroke.

For convenience an approximate length of cord is first found and the cord cut in two parts, one attached to the reducing motion; the other to the indicator. A hook should be fastened to the free end of the piece attached to the indicator. A loop is made in the free end of the piece from the reducing motion. The hook is then attached to the loop and the extra length of cord taken up by tying knots. Another method for adjusting the length of the cord is the arrangement shown in Fig. 17. The hook A is attached to the indicator cord. The cord B from the reducing motion passes through the holes in the plate P as shown. To adjust the length of the cord it is slackened at the point B and the plate slipped along the cord.

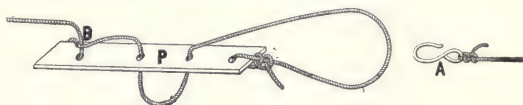


Fig. 17.

To Take Indicator Diagrams. The indicator should be in good working order before attaching to the cylinder; it should be clean, well oiled, and the levers and springs should work smoothly. The pencil point should be sharp and the pressure adjusted to make a distinct fine line.

The spring should be selected that will give a diagram $1\frac{1}{2}$ to 2 inches in height. If the spring chosen is too light, the lines are likely to be wavy from the vibration of the pencil levers.

When the indicator is in position a satisfactory reducing motion obtained and the cord adjusted, the paper should be wrapped smoothly around the drum. The edges projecting over the clips should be folded back so that they will not touch the pencil lever. Before taking the card, allow the steam to enter the indicator and move the piston up and down until the parts have become thoroughly warmed. Then pass the pencil against the paper long enough to take the diagram. Some engineers allow the pencil to remain in contact with the paper during but one revolution of the engine; others trace the diagram two or

more times. After the diagram is taken the cock is shut and without unhooking the cord the pencil is again pressed to the paper to take the atmospheric line. The cord is then disconnected and the card removed from the drum. The scale of spring, the dimensions and speed of the engine, the date, and all useful particulars are written on the cards.

If one indicator is used for both ends the three way cock shown in Fig. 12 is opened to admit steam from one end, the diagram taken; then opened for the other end and that diagram taken. Then the steam is shut off from both ends and the atmospheric line taken.

As has been said before, the indicator is of great importance to the engineer. It is used to find the indicated horse-power of the engine, and by comparison of the indicated horse-power with the brake horse-power, the **mechanical efficiency** is obtained. The indicator card shows several other things; the *time* and *manner* of the **four events** of the stroke, namely, the admission, cut-off, release, and compression. These four events make up what is called the **steam distribution**. It shows faults in the setting and working of the valves.

We have seen how the indicator diagram represents the net work done on the piston in one stroke.

Work is equal to pressure multiplied by the distance through which it acts. The *distance* is the length of stroke multiplied by the number of strokes per minute. The *pressure* is the *average net* pressure acting on the piston during the stroke. This average net pressure is called the **mean effective pressure**. If we know from the indicator card the mean effective pressure per square inch, we can find the total pressure by multiplying it by the area of the piston in square inches.

The distance per minute is equal to the length of stroke multiplied by the number of strokes.

Let P = mean effective pressure in pounds per square inch.

A = area of piston in square inches.

L = length of stroke in feet.

N = number of strokes per minute.

Then the work done per minute,

$$W = P L A N.$$

Since one horse-power is the rate of doing work when 33,000 foot-pounds of work are done per minute the **indicated horse-power** of an engine is obtained by means of the formula,

$$\text{H. P.} = \frac{P L A N}{33,000}$$

The length of the stroke, the area of the piston and the number of strokes are easily found. Then all that remains to be determined before the horse-power is calculated is the value of P , or the mean effective pressure.

Suppose one side of the piston is in communication with the boiler during the entire stroke, the mean pressure is then the boiler pressure. But if the supply of steam is cut off before the stroke is completed, the mean pressure will not equal the boiler pressure. In Fig. 1 we saw that the area of the shaded portion was equal to the length multiplied by the height. The area of a figure of any shape can be reduced to that of a rectangle having a length equal to the extreme length of the figure. Then whenever we know the area and length, we can find the height or mean height by dividing the area by the length.

Suppose a diagram like that shown in Fig. 2 has an area of $5\frac{1}{4}$ square inches and its extreme length is $3\frac{1}{2}$ inches; then the height is 5.25 divided by $3.5 = 1.5$ inches. Then with any indicator card the area of the diagram is equal to the area of a rectangle, the length of which is known and the height can easily be computed.

Suppose that we have taken a card and know that the mean height is $1\frac{1}{2}$ inches. In order to find the horse-power we must reduce the $1\frac{1}{2}$ inches to pounds pressure. If we multiply the height by the strength of the spring we get the desired result. For instance, if we had used a 30 pound spring (that is one which causes the pencil of the indicator to move one inch for every 30 pounds pressure in the cylinder) the height $1\frac{1}{2}$ inches would equal $30 \times 1\frac{1}{2} = 45$ pounds pressure.

Example. An indicator card has an area of 1.925 square inches and is 2.2 inches long. If a 60 pound spring is used what is the mean pressure?

The mean height equals $\frac{1.925}{2.2} = .875$ inch and

$$.875 \times 60 = 52.5 \text{ pounds.}$$

Suppose the engine from which the above card was taken had a piston 14 inches in diameter and a stroke of 24 inches. If it were running at 150 revolutions per minute what is its horse-power? Assume the mean effective pressure to be the same for both sides of the piston.

$$\begin{aligned} \text{H. P.} &= \frac{P L A N}{33,000} \\ &= \frac{52.5 \times 2 \times 153.94 \times 300}{33,000} \\ &= 147. \text{ (about)} \end{aligned}$$

In most engines more work is done at one end of the cylinder than at the other; it is not safe then to assume the mean effective pressure of one side the same as that of the other. Cards should be taken from each end and calculated for mean effective pressure separately, then averaged. Also the area of one side of the piston is greater than the other on account of the piston-rod. The two ends may be figured separately or the average area of the two sides of the piston may be used as the value of A .

Another method is to find the work done at each end of the cylinder and then add the results. This enables the engineer to know if his valves are set so that each end does about the same amount of work.

An engine has the following dimensions. The piston is 12 inches in diameter, the piston-rod is $2\frac{1}{8}$ inches in diameter and the length of stroke is 34 inches. While running at 92 revolutions cards were taken. The area of the card from the head end was 5.36 square inches, that of the crank end was 5.30 square inches, and a 40 pound spring was used. The cards were 3.72 inches long. We wish to know what horse-power the engine developed and which end was doing the most work.

The area of the piston is 113.097 square inches for the head end and $113.097 - 3.5466 = 109.55$ square inches for the crank end.

Then for the head end,

$$\text{H. P.} = \frac{P L A N}{33,000} = \frac{P \times 34 \times 113.097 \times 92}{12 \times 33,000} = .8933 P$$

and for the crank end,

$$\text{H. P.} = \frac{P L A N}{33,000} = \frac{P \times 34 \times 109.55 \times 92}{12 \times 33,000} = .8653 P$$

The value of P is found from the cards. Since the cards were 3.72 inches long and the area of the card from the head end was 5.36 square inches the mean ordinate is $\frac{5.36}{3.72} = 1.44$ inches which equals $1.44 \times 40 = 57.6$ pounds. The horse-power would be,

$$.8933 \times 57.6 = 51.45$$

For the crank end the mean effective pressure is found as before,

$$\frac{5.30}{3.72} = 1.425 \text{ and } 1.425 \times 40 = 57.$$

The horse-power would be,

$$.8653 \times 57 = 49.32$$

The horse-power is evidently the sum of these two quantities or $51.45 + 49.32 = 100.77$.

The head end is doing more work than the crank end but the difference is slight being only,

$$51.45 - 49.32 = 2.13 \text{ horse-power,}$$

or about 2.1 per cent. of the total power.

Considerable arithmetical work is necessary when the I. H. P. is found from the formula,

$$\text{I. H. P.} = \frac{\text{PLAN}}{33,000}$$

and the chances for error are of course, great. To save time and reduce the chance for error a table of engine constants has been prepared. The number of strokes, or twice the number of revolutions, multiplied by the length of stroke in feet is called the

piston speed. Then in the formula $\text{I. H. P.} = \frac{\text{PLAN}}{33,000}$, L = piston speed in feet per minute. In the following table, the I. H. P. of an engine is easily computed by multiplying the constant, corresponding to the diameter of the piston, by the piston speed and by the M. E. P. Or, in other words, the constants in the table equal the horse-power for an engine with a given diameter of piston having a piston speed of one foot per minute and a M. E. P. of one pound.

TABLE OF ENGINE CONSTANTS.

Diam- eter of Cylin- der.	Even Inches.	$\frac{1}{2}$ or .125.	$\frac{3}{4}$ or .25.	$\frac{1}{2}$ or .375.	$\frac{1}{2}$ or .5.	$\frac{1}{2}$ or .625.	$\frac{1}{2}$ or .75.	$\frac{1}{2}$ or .875.
1	.0000238	.0000301	.0000372	.0000450	.0000535	.0000628	.0000729	.0000837
2	.0000952	.0001074	.0001205	.0001342	.0001487	.0001640	.0001800	.0001967
3	.0002142	.0002324	.0002514	.0002711	.0002915	.0003127	.0003347	.0003574
4	.0003808	.0004050	.0004299	.0004554	.0004819	.0005091	.0005370	.0005656
5	.0005950	.0006251	.0006560	.0006876	.0007199	.0007530	.0007869	.0008215
6	.0008568	.0008929	.0009297	.0009672	.0010055	.0010445	.0010844	.0011249
7	.0011662	.0012082	.0012510	.0012944	.0013387	.0013837	.0014295	.0014759
8	.0015232	.0015711	.0016198	.0016693	.0017195	.0017705	.0018222	.0018746
9	.0019278	.0019817	.0020363	.0020916	.0021479	.0022048	.0022625	.0023209
10	.0023800	.0024398	.0025004	.0025618	.0026239	.0026867	.0027502	.0028147
11	.0028798	.0029456	.0030121	.0030794	.0031475	.0032163	.0032859	.0033561
12	.0034272	.0034990	.0035714	.0036447	.0037187	.0037934	.0038690	.0039452
13	.0040222	.0040999	.0041783	.0042576	.0043375	.0044182	.0044997	.0045819
14	.0046648	.0047484	.0048328	.0049181	.0050039	.0050906	.0051780	.0052661
15	.0053550	.0054446	.0055349	.0056261	.0057179	.0058105	.0059039	.0059979
16	.0060928	.0061884	.0062847	.0063817	.0064795	.0065780	.0066774	.0067774
17	.0068782	.0069797	.0070819	.0071850	.0072887	.0073932	.0074985	.0076044
18	.0077112	.0078187	.0079268	.0080360	.0081452	.0082560	.0083672	.0084791
19	.0085918	.0087052	.0088193	.0089343	.0090499	.0091663	.0092835	.0094013
20	.0095200	.0096393	.0097594	.0098803	.0100019	.0101243	.0102474	.0103712
21	.0104958	.0106211	.0107472	.0108739	.0110015	.0111299	.0112589	.0113886
22	.0115192	.0116505	.0117825	.0119152	.0120487	.0121830	.0123179	.0124537
23	.0125902	.0127274	.0128654	.0130040	.0131435	.0132837	.0134247	.0135664
24	.0137088	.0138519	.0139959	.0141405	.0142859	.0144321	.0145789	.0147266
25	.0148750	.0150241	.0151739	.0153246	.0154759	.0156280	.0157809	.0159345
26	.0160888	.0162439	.0163997	.0165563	.0167135	.0168716	.0170304	.0171899
27	.0173502	.0175112	.0176729	.0178355	.0179988	.0181627	.0183275	.0184929
28	.0186592	.0188262	.0189939	.0191624	.0193316	.0195015	.0196722	.0198436
29	.0200158	.0201887	.0203624	.0205368	.0207119	.0208879	.0210645	.0212418
30	.0214200	.0215988	.0217785	.0219588	.0221399	.0223218	.0225044	.0226877
31	.0228718	.0230566	.0232422	.0234285	.0236155	.0238033	.0239919	.0241812
32	.0243712	.0245619	.0247535	.0249457	.0251387	.0253325	.0255269	.0257220
33	.0259182	.0261149	.0263124	.0265106	.0267095	.0269092	.0271097	.0273109
34	.0275128	.0277155	.0279189	.0281231	.0283279	.0285336	.0287399	.0289471
35	.0291550	.0293636	.0295729	.0297831	.0299939	.0302056	.0304179	.0306309
36	.0308448	.0310594	.0312747	.0314908	.0317075	.0319251	.0321434	.0323624
37	.0325822	.0328027	.0330239	.0332460	.0334687	.0336922	.0339165	.0341415
38	.0343678	.0345937	.0348209	.0350489	.0352775	.0355070	.0357372	.0359681
39	.0361998	.0364322	.0366654	.0368993	.0371339	.0373694	.0376055	.0378424
40	.0380800	.0383184	.0385575	.0387973	.0390379	.0392793	.0395214	.0397642
41	.0400078	.0402521	.0404972	.0407430	.0409895	.0412368	.0414849	.0417337
42	.0419832	.0422335	.0424845	.0427362	.0429887	.0432420	.0434959	.0437507
43	.0440062	.0442624	.0445194	.0447771	.0450355	.0452947	.0455545	.0458151
44	.0460768	.0463389	.0466019	.0468655	.0471299	.0473951	.0476609	.0479276
45	.0481950	.0484631	.0487320	.0490016	.0492719	.0495430	.0498149	.0500875
46	.0503608	.0506349	.0509097	.0511853	.0514615	.0517386	.0520164	.0522949
47	.0525742	.0528542	.0531349	.0534165	.0536988	.0539818	.0542655	.0545499
48	.0548352	.0551212	.0554079	.0556953	.0559835	.0562725	.0565622	.0568526
49	.0571438	.0574357	.0577284	.0580218	.0583159	.0586109	.0589065	.0592029
50	.0595000	.0597979	.0600965	.0603959	.0606959	.0609969	.0612984	.0616007
51	.0619038	.0622076	.0625122	.0628175	.0631235	.0634301	.0637379	.0640462
52	.0643552	.0646649	.0649753	.0652867	.0655987	.0659115	.0662250	.0665392
53	.0668542	.0671699	.0674864	.0678036	.0681215	.0684402	.0687597	.0690799
54	.0694008	.0697225	.0700449	.0703681	.0706923	.0710166	.0713419	.0716681
55	.0719950	.0724226	.0728510	.0732801	.0737099	.0741406	.0745719	.0749309
56	.0746368	.0749704	.0753047	.0756398	.0759755	.0763120	.0766494	.0769874
57	.0773262	.0776657	.0780060	.0783476	.0786887	.0790312	.0793745	.0797185
58	.0800632	.0804087	.0807543	.0811019	.0814495	.0817980	.0821472	.0824971
59	.0828478	.0831992	.0835514	.0839043	.0842579	.0846123	.0849675	.0853234
60	.0856800	.0860374	.0863955	.0867543	.0871139	.0874743	.0878354	.0881973

To Use the Table. If the diameter of the piston is an even number, the constant is found in the second column; if it contains a fraction the constant is found by following the column horizontally until the required fraction is reached. The constant multiplied by the piston speed in feet per minute and by the M. E. P. in pounds per square inch gives the I. H. P.

Example. An engine runs at 75 revolutions. The stroke is 4 feet; if the M. E. P. is 48 pounds and the piston $27\frac{3}{8}$ inches in diameter what is the I. H. P.

From the table the constant for a piston $27\frac{3}{8}$ inches in diameter is .0178355. The piston speed is $150 \times 4 = 600$ feet per minute. Then the I. H. P. is,

$$.0178355 \times 600 \times 48 = 513.66$$

The horse-power as above calculated is called the indicated horse-power and is usually written I. H. P. Although the above calculation shows the amount of power the engine develops it does

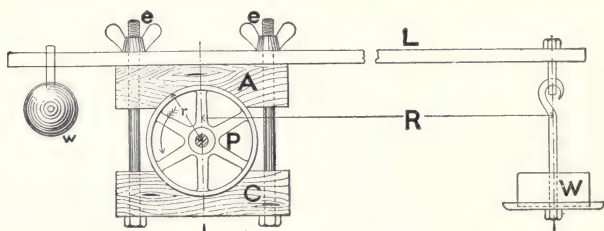


Fig. 18.

not show the *available* power since part of the indicated horse-power is used to run the engine itself, that is, to overcome the friction of the parts. To determine how much power can be used to run machinery some form of absorption dynamometer or friction brake is attached to the engine. The power thus obtained is called the **Brake Horse Power** or B. H. P. It is more satisfactory for both the owner and builder to know the B. H. P. than to know the I. H. P.

The Prony Brake, Fig. 18, is one of the simplest absorption dynamometers. The two wooden blocks A and C are held together against the rim of the pulley P by bolts. The thumb-nuts, e, e, being

used to adjust the pressure. By means of the bolts the arm L is held to the upper block. From this arm is suspended the ball weight, w , which by sliding along the arm counterbalances the weight of the arm and pan at the other end. The pulley revolves at the required speed in the direction indicated by the arrow. The bolts are tightened until the lever remains stationary in a horizontal position when a known weight, W , is hung at the end.

The amount of work absorbed by the brake depends upon the weight W , the length, R , and the speed. It is independent of the diameter of the pulley and the pressure of the block because the moments of force about the center of the pulley are equal when the lever L , is horizontal. Letting f equal the co-efficient of friction, p the pressure of the blocks and r the radius of the pulley,

$$f p r = W R$$

The work done at the face of the pulley equals the force multiplied by the distance or the pressure multiplied by the number of feet passed through.

Let N = the number of revolutions per minute. Then the distance passed through per minute equals $2 \pi r N$ and the work done equals $2 \pi r N f p$. Then as $f p r = W R$, the work done at the rim of the pulley equals the left hand side of the equation multiplied by $2 \pi N$, and to keep both sides equal we multiply $W R$ by $2 \pi N$. Hence the work done is expressed by the formula

$$\begin{aligned} 2 \pi N W R \text{ and,} \\ \text{B. H. P.} &= \frac{2 \pi N W R}{33,000} \\ &= .0001904 N W R \end{aligned}$$

A Prony brake with an arm 4 feet long was attached to the pulley on the fly wheel of an engine. The weight in the scale pan was 50 pounds and the speed of the engine 300 revolutions. Find the brake horse power.

$$\begin{aligned} \text{B. H. P.} &= .0001904 \times 300 \times 50 \times 4 \\ &= 11.424 \end{aligned}$$

The **rope brake** shown in Fig. 19 is easily constructed of material at hand and being self-adjusting needs no accurate fitting. For large powers the number of ropes may be increased. It is considered a most convenient and reliable brake. In Fig. 19 the spring

balance, B, is shown in a horizontal position. This is not at all necessary; if convenient the vertical position may be used. The ropes are held to the pulley or fly-wheel face by blocks of wood, O. The weights at W may be replaced by a spring balance if desirable.

To calculate the Brake Horse Power, subtract the pull registered by the spring balance, B, from the load at W. The lever arm is the radius of the pulley plus $\frac{1}{2}$ the diameter of the rope. The formula is,

$$\begin{aligned} \text{B. H. P.} &= \frac{2 \pi R N (W - B)}{33,000} \\ &= .0001904 R N (W - B) * \end{aligned}$$

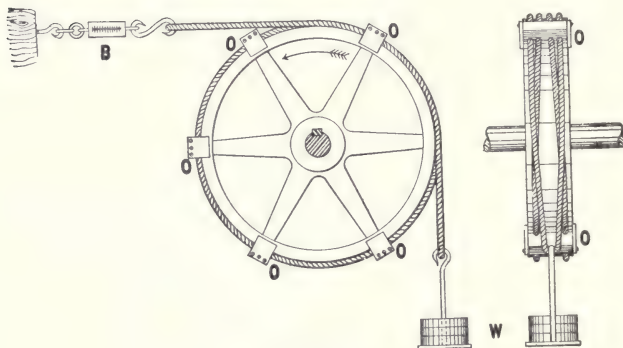


Fig. 19.

Example. A rope brake is attached to a gas engine. The average reading of the spring balance is 8 pounds. $W = 80$ pounds. If the radius of the brake wheel is 28 inches and the rope 1 inch in diameter, what is the B. H. P. when the engine makes 350 revolutions per minute?

$$\begin{aligned} R &= 28 + \frac{1}{2} = 28\frac{1}{2} \text{ inches} = \frac{28.5}{12} \text{ feet} \\ \text{B. H. P.} &= .0001904 R N (W - B) \\ &= .0001904 \times \frac{28.5}{12} \times 72 \times 350 \\ &= 11.4 \text{ Ans} \end{aligned}$$

If both the indicated horse-power and the brake horse-power

* NOTE: If B is greater than W, the engine is running in the opposite direction. Use the formula B. H. P. = .0001904 R N (B - W).

are known the power used in friction is found by subtracting the B. H. P. from the I. H. P.

The **mechanical efficiency** of the engine is the **ratio** of the B. H. P. to the I. H. P. or,

$$E = \frac{\text{B. H. P.}}{\text{I. H. P.}}$$

If an engine of 18.2 indicated horse-power develops at a trial 16.02 brake horse-power, what is its mechanical efficiency?

$$\begin{aligned} E &= \frac{\text{B. H. P.}}{\text{I. H. P.}} \\ &= \frac{16.02}{18.2} = .88 \\ &= 88 \% \text{ Efficiency.} \end{aligned}$$

Brakes should be well lubricated. For small powers the heat generated by friction between the ropes or blocks and the rim of the wheel, will be conducted away by radiation but for large powers some additional means is necessary. In case there are flanges on the wheel, water can be introduced into the wheel, the flanges keeping it from flowing out and centrifugal force keeping it in contact with the rim. The amount of water can be regulated so that all may be evaporated, or a scoop can be arranged to carry off the water. In all cases the water should flow continuously.

To Find the Area of Cards. M. E. P. or the mean effective pressure is equal to the area of the indicator diagram divided by the length. The length is easily found by measurement but to find the area is more difficult since the shape is irregular. If the figure were regular its area could be found by geometry or by simple formulas.

The area of the indicator card can be found in two ways. By dividing the diagram into sections and by the use of a planimeter. The former is only an approximate method; the area thus found is nearly correct if the number of divisions is great.

Tangents at each end, perpendicular to the atmospheric line are first drawn. The horizontal distance between these tangents is then divided into 10 or more equal parts. The horizontal length of each section is then divided into two equal parts and lines perpendicular to the atmospheric line drawn through these points of

division. The sum of the lengths of all these lines is divided by the number of lines to get the average. This average length or average ordinate multiplied by the scale of spring gives the mean effective pressure.

Fig. 20 is the card from the crank end of an engine. The

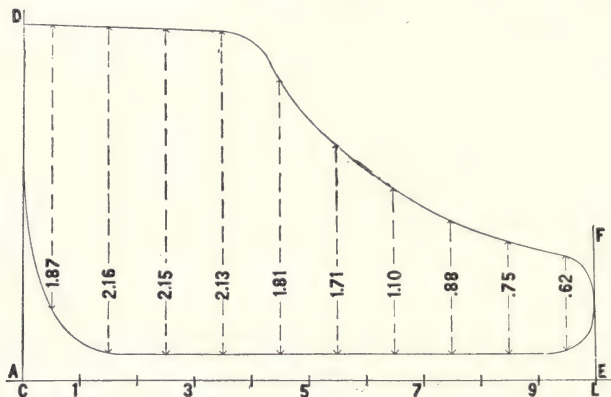


Fig. 20.

line C L is the atmospheric line and the lines A D and E F are drawn perpendicular to it and tangent to the extreme ends of the diagram. The line A E is divided into 10 equal parts and lines

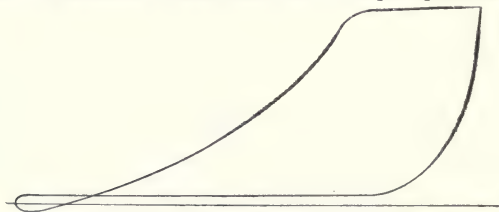


Fig. 21.

are drawn through points marking the centers of the divisions. On each of these lines the length is marked. The sum of the lengths is 15.18 and 15.18 divided by 10 is 1.518. If the scale of spring is 40 pounds, 1.518 multiplied by 40 is the M. E. P. or 60.7 = M. E. P.

The horizontal length may be divided into any number of equal parts but 10 or 20 makes the computation easy. The operation of finding the M. E. P. for the head end is exactly the same. The average M. E. P. for one revolution of the engine is the average of the two mean effective pressures.

In case the diagram is very irregular it should be divided into 20 equal parts instead of 10. If there is a loop in the diagram as shown in Fig. 21 the area of the loop must be subtracted

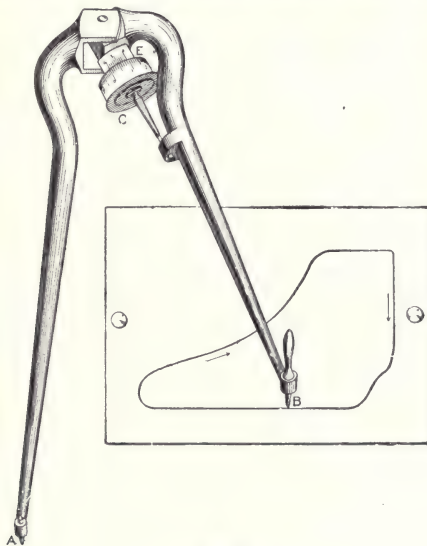


Fig. 22.

from the area of the other part as it represents work done by the piston on the steam and therefore loss.

The lengths may be marked off on a piece of paper if a good scale is not at hand.

A more accurate result is obtained by using an instrument called the **planimeter**. There are several planimeters and averaging instruments in common use for determining the mean effective pressure of indicator cards. The planimeter shown in Fig.

22 is one of the most simple and is called the Amsler Polar Planimeter from its inventor Prof. Amsler. The cut is about one-half the size of the instrument. It consists of two arms free to move about a pivot and a roller graduated in inches and tenths of inches. A vernier is placed with the roller so the areas may be read in hundredths of a square inch. The point A is kept stationary and the tracer B is moved once around the outline of the diagram. The area in square inches of the diagram is read from the roller C and the vernier E.

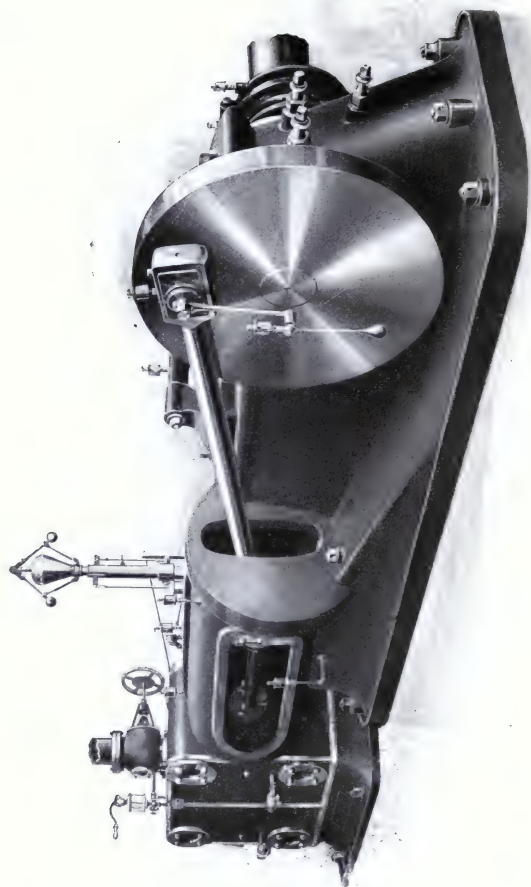
To Use the Planimeter. The diagram should be fastened to some flat unglazed surface, such as a drawing board, by means of thumb tacks, springs or pins. The point A is pressed into the paper so that it will hold in place. The point B is set at any point in the outline of the diagram and the roller set at zero. Follow the outline of the diagram carefully in the direction of the hands of a watch as indicated by the arrows in Fig. 22 until the tracer has moved completely around the diagram. The result is then read to hundredths of an inch from the roller. Suppose after tracing over the outline we find that the largest figure that has passed the zero of the vernier is 3; the number of graduations (tenths) that have passed the zero to be 5 and the number (hundredths) of the graduations in the roller that exactly coincides with a graduation on the vernier to be 9. Then the area is 3.59 square inches.

Often at the start the roller is not adjusted so that the zeros coincide but the reading is taken and subtracted from the final reading. Thus if the first reading is 4.63 and the second 7.31 the area is $7.31 - 4.63 = 2.68$ square inches. In case the second reading is less than the first, add 10 to the second reading then subtract.

This instrument is very valuable to an engineer who takes indicator cards. The results obtained are very accurate, the error being small. Ten or twelve diagrams can be measured by this instrument in the same time that is necessary to measure a single card by the method of ordinates.

It is well to run over the area three or four times and take an average as the tracing of the diagram cannot be absolutely correct at any time.





MURRAY ROLLING MILL TYPE CORLISS ENGINE FOR HIGH PRESSURES AND HIGH SPEEDS

THERMAL EFFICIENCY.

The **thermal efficiency** of the steam engine is found in the same manner as that of any other heat engine. The efficiency depends upon the limits of temperature and not upon the nature of the working medium.

Let T_1 = absolute temperature of the heat received by the engine.

T_2 = absolute temperature of the heat rejected by the engine.

E = efficiency of engine.

Then,

$$E = \frac{T_1 - T_2}{T_1}$$

or, the efficiency equals the temperature of the heat rejected, subtracted from the temperature of the heat received and the result divided by the temperature of the heat received.

Suppose an engine is supplied with steam at 120 pounds absolute pressure and the exhaust is atmospheric pressure. What is the efficiency?

The absolute temperature corresponding to 120 pounds absolute pressure is $341.05^\circ + 461^\circ$ and the temperature of atmospheric pressure is $212^\circ + 461^\circ$.

Then,

$$E = \frac{802.05 - 673}{802.05} = .16 \text{ or } 16 \text{ per cent.}$$

If the engine had been of the condensing type and the exhaust pressure one pound above the vacuum, the efficiency would be as follows:

The temperature of one pound absolute pressure is $101.99^\circ + 461^\circ$.

$$E = \frac{802.05 - 562.99}{802.05} = .30 \text{ or } 30 \text{ per cent.}$$

In actual engines this efficiency cannot be obtained because the difference between the amount of heat received and that rejected is not all converted into work. Part of it is lost by radiation, conduction, leakage, etc. Also cylinder condensation reduces the efficiency.

The Theoretical Indicator Diagram. An indicator diagram is the result of two movements; a horizontal movement of the

paper and a vertical movement of the pencil. The horizontal movement exactly corresponds to the movement of the piston of the engine and the vertical movement exactly corresponds to the pressure of steam in the cylinder.

The shape of the indicator card depends upon the manner in which steam is admitted to and released from the cylinder. Different engines give different shaped indicator cards and the cards taken from an engine vary with the conditions. Figs. 1 and 2 show theoretical indicator cards from a non-condensing engine without clearance; the former being for the case that has admission during the whole stroke. The diagram of Fig. 2 shows the cut off at $\frac{1}{2}$ stroke. All practical engines have clearance and slight compression; so the theoretical diagram assumes the shape

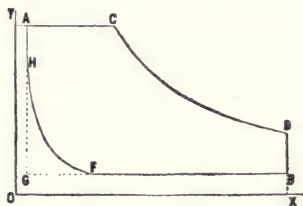


Fig. 23.

shown in Fig. 23. In this card the admission line HA is vertical, the steam line AC is horizontal, the expansion line CD an hyperbolic curve, the exhaust line DB vertical, the back pressure line BF horizontal and the compression curve an hyperbola. The **actual** shape is somewhat different from the **theoretical**

mainly because the valves do not open and close quickly, the ports offer some resistance to the passage of the steam and the back pressure is neither atmospheric in the non-condensing engine nor absolute vacuum in the condensing engine.

The diagram shown in Fig. 24 is a practical diagram and like those taken from engines.

The **atmospheric line** LM is the line drawn by the pencil of the indicator when the connection to the engine is closed and both sides of the piston of the indicator are open to the atmosphere. It is the zero of the steam gage.

The **admission line** HA shows the rise of pressure due to the admission of steam to the cylinder. If the steam is admitted quickly when the engine is nearly on dead center this line will be very nearly vertical.

The **steam line** AC is drawn while the valve admits steam

to the cylinder. This line is horizontal if there is no wire-drawing.

The **point of cut off C**, indicates the point at which the admission of steam is stopped by the closing of the valve. This point is rounding since the valve closes slowly. Sometimes it is difficult to determine the exact point where cut off takes place; it is usually where the curve changes from concave to convex.

The **expansion curve C D** shows the fall in pressure as the steam expands while the piston moves toward the end of the stroke.

The **point of release D** shows the point at which the exhaust

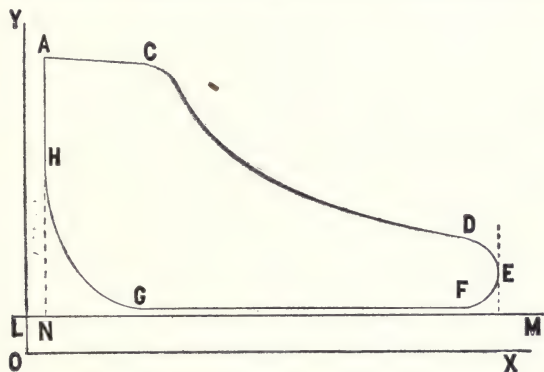


Fig. 24.

valve opens. The rounding is due to the slow action of the valve when opening. Because of this slow action of the valve, release begins a little before the end of the forward stroke.

The **exhaust line D E F** represents the loss in pressure which occurs while the valve opens to exhaust at and near the end of the stroke.

The **back pressure line F G** shows the back pressure against which the piston acts during the return stroke. For a condensing engine this line is below the atmospheric line **L M**, the distance below being dependent upon the state of the vacuum in the condenser. For cards taken from a non-condensing engine the back pressure line is a little above the atmospheric line.

The **point of exhaust closure G** is the point where the valve

closes to exhaust. The exact point is not clearly defined as the curve shows a change of pressure due to the gradual closing of the valve.

The **compression curve** G H shows the rise of pressure due to the compression of the steam remaining in the cylinder after the valve has closed to exhaust.

The **zero line** of pressure or line of absolute vacuum O X is drawn below and parallel to the atmospheric line. The distance between the lines O X and L M represents 14.7 pounds pressure.

The **clearance line** O Y is drawn perpendicular to the line of absolute vacuum and at a distance from the end of the diagram

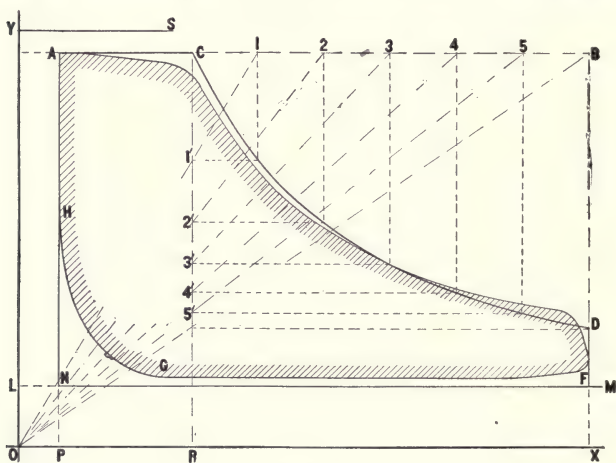


Fig. 25.

equal to the same per cent. of the length of the diagram as the clearance volume is of the piston displacement, or

$$\frac{L N}{L M} = \frac{\text{clearance volume}}{\text{volume of cylinder}}.$$

It is readily seen that the area of an actual indicator diagram is less than that of a theoretical card. This is because of the round corners at cut off and exhaust, the back pressure and the compression. Sometimes it is useful, especially in designing engines, to draw the theoretical indicator card.

To Draw the Theoretical Card. To draw an ideal diagram draw P X equal to the length of stroke and O P equal to the clearance shown in Fig. 25. Draw O Y and P A perpendicular to O X and draw Y S parallel to O X and at a height corresponding to the boiler pressure.

The line of initial pressure A C is then drawn parallel to Y S and is usually taken as from 90 to 95 per cent. of the boiler pressure if there is no special cause for loss. Then take A C as the portion of the stroke at which steam is admitted so that $\frac{P X}{A C} =$

the ratio of expansion. The expansion line is considered a hyperbolic curve with O Y and O X as asymptotes. To draw the hyperbolic curve. First draw the line A C B parallel to the atmospheric line and F D B and R C perpendicular to it. Then make points 1, 2, 3, 4, etc., on C B and connect them with the point O. At the points 1', 2', 3', 4', etc., where these lines intersect the line R C draw parallels to C B until they meet perpendiculars from 1, 2, 3, 4. The point of intersection of these lines are points in the hyperbolic curve C D, as shown in Fig. 25. Any number of points may be used; but there must be enough to determine the curve.

The area A C D F N H is the theoretical card, with a given boiler pressure, ratio of expansion and an assumed back pressure. The actual card for the same data would be more nearly like the shaded area which lies mostly within the outline of the theoretical card. In designing engines it is well to know the ratio of the actual to the ideal card for all types of engines.

This ratio varies with the speed, type of engine, and kind of valves and has the following values.

For ordinary plain slide valve engines,	.8 to .9
For small engines, about,	.85
For large engines, about,	.90
For engines with high speed, about,	.75
For compound engines,	.75 to .80
For compound engines, high speed,	.65 to .75
For triple expansion engines,	.60 to .70
For triple expansion engines, high speed,	.50 to .60

CARDS FOR COMPOUND ENGINES.

In Fig. 26 the ideal cards of a **tandem** compound engine of the Woolf type are shown. The diagram A C D E F G H is from the high pressure cylinder and the diagram L M N P Q R from the low pressure. S T is the atmospheric line.

The line H A is the admission line, A C the steam line, and C D the expansion line. These lines are similar to those of a single cylinder engine diagram. At D, the point of release, the pressure drops slightly as steam is admitted to the low pressure

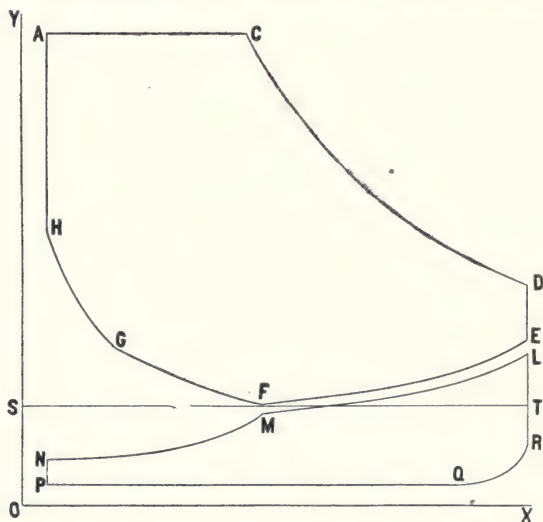


Fig. 26.

cylinder. The back pressure line E F, of the high pressure cylinder is parallel to the steam line L M, of the low pressure cylinder. They would coincide if there were no resistance to the flow of steam and no heat loss. The flow of steam from the high to the low pressure cylinder is cut off at F and the steam remaining in the high pressure cylinder is compressed in the cylinder and in the pipes. The exhaust closes at G and steam is compressed in the small cylinder from G to H.

Cut off occurs in the low pressure cylinder at M, and the steam in the cylinder expands, the curve M N being an equilateral hyperbola. Release takes place at N and the pressure falls to that of the condenser. The back pressure line P Q and the compression curve Q R are like those of a single engine.

The diagrams shown in Fig. 27 are from a Woolf engine. The lines are similar to those of the theoretical diagram shown in Fig. 26.

With compound engines of the Woolf type the steam passes

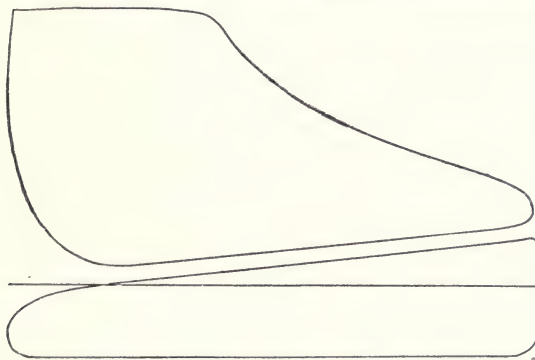


Fig. 27.

directly from the high pressure cylinder to the low pressure. In the **cross** compound, the cranks being at 90° , the piston in one cylinder is at the *end* of the stroke when the other is at the *middle* of the stroke. Therefore, a **receiver** must be used. The ideal cards from a cross compound engine are shown in Fig. 28. The dimensions are as follows:

Volume of high pressure cylinder	=	5 cu. ft.
Volume of receiver	=	5 cu. ft.
Volume of low pressure cylinder	=	15 cu. ft.
Cut off of high pressure cylinder	=	$\frac{1}{4}$ stroke.
Cut off of low pressure cylinder	=	$\frac{1}{2}$ stroke.
Initial pressure (absolute)	=	120 lbs.
Back pressure (absolute)	=	3 lbs.

Clearance and compression are not considered. The stroke of the high pressure cylinder begins at A. Steam is cut off at $\frac{1}{4}$ stroke at C.

Since steam is cut off at $\frac{1}{4}$ stroke, the ratio of expansion in the high pressure cylinder is 4. Then the volume of steam at D is 4 times that at C and since $p v = p_1 v_1$, the pressure at D is $\frac{120 \times 1}{4} = 30$ pounds. Steam is released at D and passes to the receiver. The pressure at release, L, of the low pressure cylinder is found from the equation $p v = p_1 v_1$. The volume of steam in the high pressure cylinder at cut off, is $\frac{5}{4}$ cubic feet and the

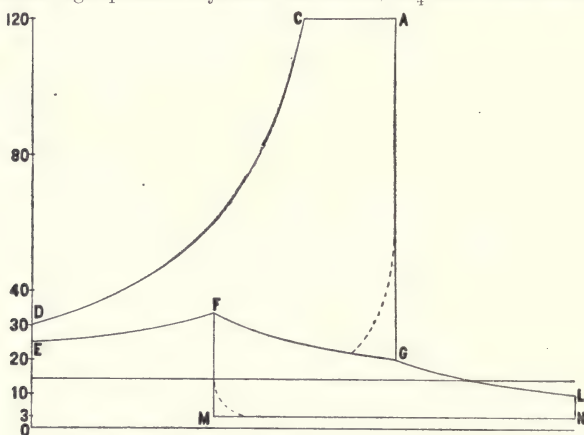


Fig. 28.

volume at L is 15 cubic feet; then the total ratio of expansion is $15 \div \frac{5}{4} = 12$. Then the pressure at L is $\frac{120 \times 1}{12} = 10$ pounds.

Since the low pressure cylinder cuts off at $\frac{1}{2}$ the stroke the volume at G is $\frac{1}{2}$ that at L.

Then as $p v = p_1 v_1$, $p \times \frac{1}{2} = 10 \times 1$, or $p = 20$ pounds. Hence the pressure at G is 20 pounds.

We can now find the pressure at E. The pressure in the receiver when the low pressure cylinder cut off, was 20 pounds because the low pressure cylinder was in communication with it,

Then as the piston in the small cylinder is at the end of the stroke when the steam is cut off in the large cylinder, steam at 30 pounds pressure rushes from the high pressure cylinder into the receiver and mixes with steam at 20 pounds pressure. When steam is cut off in the low pressure cylinder the high pressure cylinder and the receiver are in communication with each other; the total volume being the volume of the high pressure cylinder plus that of the receiver or $5 + 5 = 10$ cubic feet. What we wish to find is the pressure of the steam of this volume. The formula is,

$$P V = p v + p_1 v_1.$$

Since there are 10 cubic feet in the receiver and high pressure cylinder, the value of V is 10. The volume of steam in the high pressure cylinder is 5 cubic feet and its pressure is 30 pounds. The volume of steam in the receiver is 5 cubic feet and its pressure is 20 pounds, then,

$$10 P = 5 \times 30 + 5 \times 20$$

$$P = \frac{250}{10} = 25 \text{ pounds.}$$

Then the pressure at $E = 25$ pounds.

While the piston of the high pressure cylinder is on the return stroke, steam in that cylinder is compressed from E to F ; the volume being 10 cubic feet. When the piston has completed $\frac{1}{2}$ the return stroke, the volume at F is equal to $\frac{1}{2}$ the volume of the high pressure plus the volume of the receiver or $\frac{5}{2} + 5 = 7\frac{1}{2}$ cubic feet.

Then with the formula $p v = p_1 v_1$ we obtain,

$$p = \frac{25 \times 10}{7\frac{1}{2}} = 33.33$$

or the pressure at F is 33.33 pounds.

Then as the cranks are 90° apart, and the high pressure piston is at the middle of the stroke, the low pressure piston must be at the beginning of its stroke; or the pressure in the low cylinder is the same as that in the high and in the receiver, or 33.33 pounds. Since the stroke is beginning in the low pressure cylinder, steam is being admitted to it from the receiver and consequently the pressure in the large cylinder falls as the volume increases, which

as shown by the line F G. Cut-off occurs in the low pressure cylinder at G, and the steam expands until the piston reaches L; the curve being an equilateral hyperbola. At L the release occurs and the pressure drops to that in the condenser; in this case about 3 pounds. The back pressure line is of course the pressure in the condenser.

The cards of Fig. 29 are from a cross compound engine. The rise of pressure in the middle of the back pressure line of the

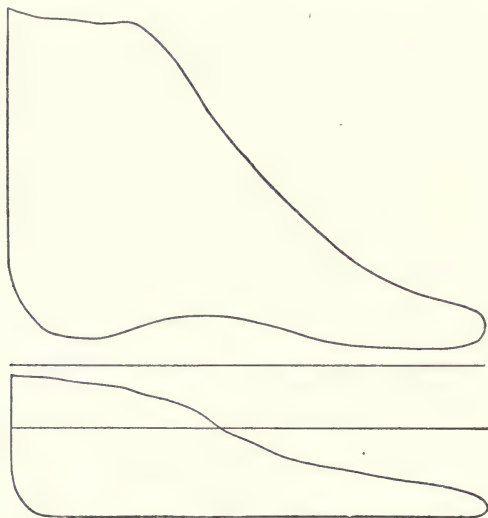


Fig. 29

diagram from the small cylinder is due to the fluctuation of pressure in the receiver.

COMBINED DIAGRAMS.

The indicator cards of multi-cylinder engines may be combined so that the pressures and volumes are shown in their proper relations. To do this, the cards are reduced to the same scale of pressure and the same scale of volume. To make the combined diagram of convenient size, the length of the low-pressure card is

left as it is and the length of the high-pressure diagram is shortened in the ratio of the cylinder volumes.

Perhaps the best way to show the method is by an illustration. The cards shown in Fig. 30 were taken from a compound condensing engine.

Ratio of cylinder volumes = 1:3
 Initial pressure = 138 pounds
 Spring, high-pressure card = 60 pounds
 Spring, low-pressure card = 30 pounds

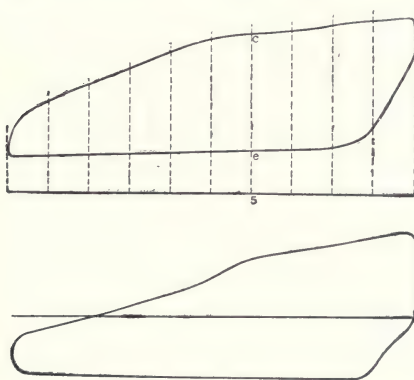


Fig. 30.

First draw the line of zero pressure $A B$ and the line of zero volume $E F$ (see Fig. 31). The line $O P$ is drawn parallel to $E F$ and at a distance proportional to the clearance of the low-pressure cylinder. Similarly the distance between $M N$ and $E F$ represents the clearance of the high-pressure cylinder. Now draw the atmospheric line $C D$. In this case it will be .49 inch above $A B$ because a 30-pound spring is used, and $\frac{14.7}{30} = .49$. Reproduce the low-pressure card without change, as shown.

Divide the high-pressure card with 10 (or more) ordinates, and reproduce it with volumes and pressures of the same scale as the low-pressure card. Since a 60-pound spring was used, each ordinate will be twice as long (because $\frac{60}{30} = 2$). The distance

between the ordinates will be $\frac{1}{3}$ as great because the high-pressure cylinder is $\frac{1}{3}$ the volume of the low. Draw the ordinates as shown. The distance between ordinates L and M will be $\frac{1}{3}$ of the length of the high-pressure card. The points *c* and *e* are on the fifth ordinate, and are twice as far above the atmospheric line as they are in the original high-pressure card.

After locating all the points draw the curve through them. Now draw the theoretical expansion curve R S through the point of cut-off of the high-pressure cylinder by the method explained

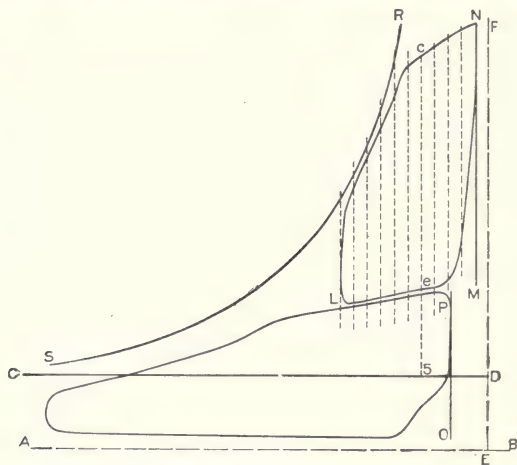


Fig. 31.

on page 39. The difference between the area included between the theoretical curve and the lines of no pressure and no volume, and the sum of the actual areas, represents approximately the losses.

This method is not strictly accurate, because all the steam used in the high-pressure cylinder does not pass to the low-pressure cylinder; a small portion is left for compression. By thus combining the cards the action of the valves may be discussed, provided such data as size, type and speed are known.

The combined diagram of a multi-expansion engine drawn

according to the above method is shown in Fig. 32. The volumes and pressures are first reduced to the scale of the low-pressure card. The cards are then redrawn at the proper distances from the lines of no pressure and no volume; the clearance in each cylinder being considered.

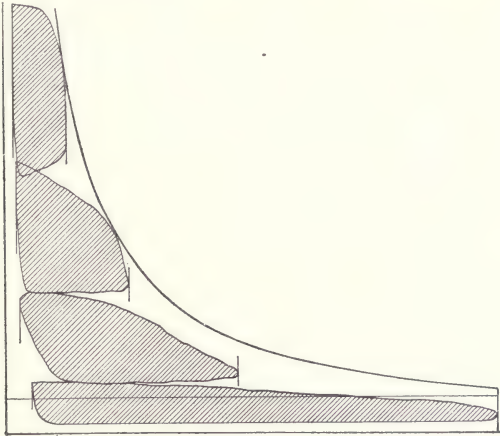


Fig. 32

HORSE-POWER OF COMPOUND ENGINES.

The I. H. P. of multi-cylinder engines may be found by adding the I. H. P. of the several cylinders. Another method is to reduce all the pressures to the area of the low-pressure cylinder. This is done by dividing the M. E. P. of each cylinder by the inverse ratio of its volume to that of the low-pressure cylinder.

Suppose the M. E. P. of the high-pressure cylinder of a compound engine is 78 pounds as found from the indicator card. If the volume of the high-pressure cylinder is $\frac{1}{3}$ that of the low, the M. E. P. of the high referred to the low would be $\frac{78}{\frac{1}{3}} = 26$ pounds.

Then if the card from the large cylinder shows a M. E. P. of 30 pounds, the total M. E. P. is $30 + 26 = 56$ pounds, and the I. H. P. is found by inserting 56 as the value of P and using the area of the low pressure as A, in the formula for I. H. P.

INDICATOR CARDS.

The indicator reveals defects in the steam distribution. That is, if the valve or valves are set so that the events of the stroke are *too early* or *too late* or if *more work* is done at one end of the cylinder than at the other, the engineer finds it out by examining the indicator card.

Sometimes an engine appears to run well and the owner is perfectly satisfied with it; but from the indicator diagrams it is found that considerable saving might be effected by correcting the defects in the valve setting.

On looking at a diagram one might say it was a faulty card

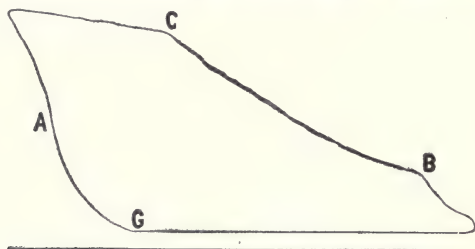


Fig. 33.

and yet for that type, size and speed it is perhaps the best that could be obtained from the engine. The same form of diagram is not possible from different types of engines. The diagram from a Corliss engine, having four valves, is different from that of the plain slide valve engine; also the diagrams from high speed engines differ from those of low speed.

The most common faults in the distribution of steam in the cylinder can be divided into four classes.

- a* = **Admission** too early or too late.
- b* = **Cut-off** too early or too late.
- c* = **Release** too early or too late.
- d* = **Compression** too early or too late.

In the following figures,

- A is the point of admission,
- C is the point of cut off,
- B is the point of release,
- G is the point of compression.

The diagram shown in Fig. 33 shows **too early admission**. The admission line curves *backward* instead of being straight and perpendicular to the atmospheric line as it is in Fig. 24. The diagram also shows cut off, release and compression early. When the valve is of the plain slide valve type *all* the events are likely to be too early if one of them is. The reason for the event being

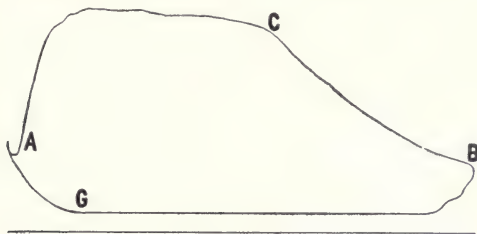


Fig. 34.

too early is that the eccentric has too much *angular advance*. Hence the remedy is to decrease the angular advance.

Fig. 34 shows a diagram having the opposite defects to those of Fig. 33. The events are too late. The admission line curves forward and the line shows that admission does not take place until after the stroke is well begun. Release occurs at the end of

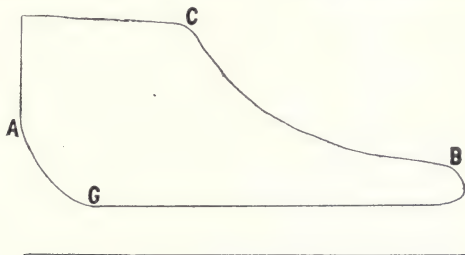


Fig. 35.

the stroke. In this case the angular advance of the eccentric should be increased until the admission line is perpendicular to the atmospheric line.

Fig. 35 shows a card having too much back pressure. This may be due to a small exhaust port or pipes, or the passage of steam

through coils of pipe for heating. The card shows the other events to be good. A diagram having **too late cut off** is shown in Fig. 36. The pressure at release is high. When the engine is running under this condition much of the benefit from expansion is lost.

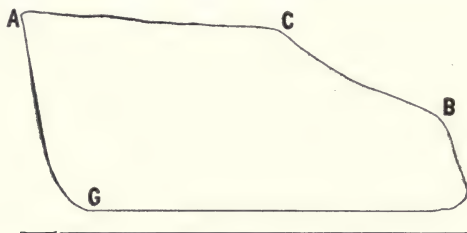


Fig. 36.

A diagram from a condensing engine is shown in Fig. 37. These oscillations are caused by the vibration of the indicator piston and spring. To avoid these vibrations never use a very light spring for high speed.

The diagram of Fig. 21 shows **too early cut off**. In this case the cut off is so early that the expansion line extends *below* the

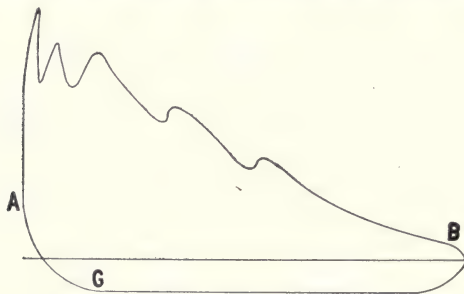
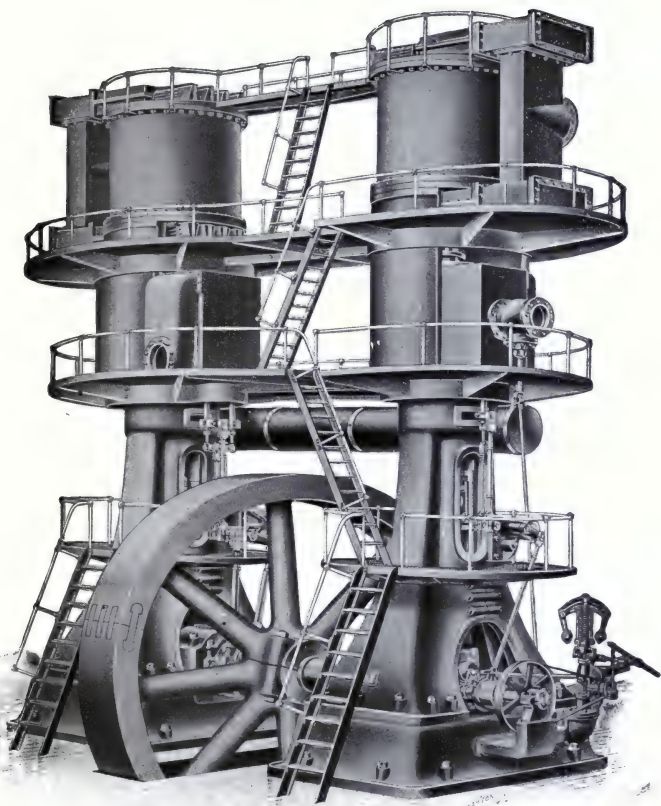


Fig. 37.

atmospheric line making a loop. The area of the loop must be *subtracted* from the area of the card as explained on page 33.

Fig. 38 shows a pair of diagrams from a plain slide valve engine. The admission lines are good. The sloping steam lines



VERTICAL CROSS COMPOUND BLOWING ENGINE.
Buckeye Engine Company.

show wire-drawing due to the slow action of the valve or too small ports or pipes. This wire-drawing decreases the area of the card which indicates loss. The greatest fault is the inequality of area of the diagram for the ends. The late cut off and consequent late compression of one end causes more area than the too early cut off and too early compression of the other end. These

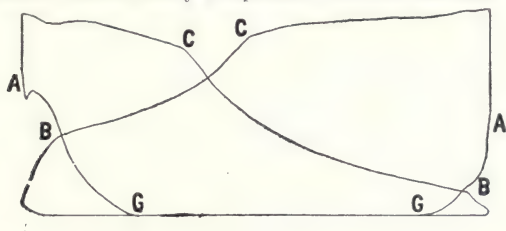


Fig. 38.

cards can be improved by adjusting the angular advance of the eccentric and the length of the valve rod.

The diagram of Fig. 39 indicates too early compression. The compression curve extends above the initial pressure line. The area of the loop must be subtracted from the card area, when com-

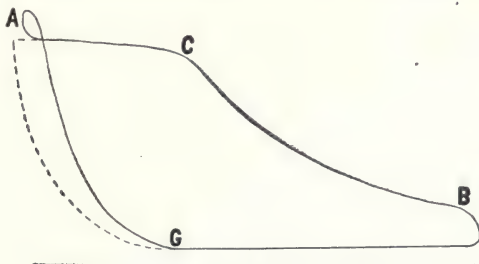


Fig. 39.

puting the I. H. P. If the cut off is kept the same and the compression made what it should be, the gain in area would be the area included between the full line and the dotted line plus the area of the loop. The remedy for this case is to decrease the inside lap.

The amount of compression varies with the speed and type of engine. Slow speed engines require *less* compression or cushioning than high speed. The exhaust steam should *never* be compressed higher than the boiler pressure.

For high speed engines the compression should extend to about .9 the initial pressure. For medium speeds about .5 and for low speed .2 to .4.

In the case of a slide valve engine it is not always possible to set the valve so that the card may have all the events as they should be. Sometimes the laps of the valves should be altered. For too much compression decreases the inside lap. For too early cut off decrease the outside lap.

If the valve travel is increased, compression is retarded, that is, decreased; release occurs sooner.

STEAM CONSUMPTION.

The amount of steam used by an engine is called its **steam consumption** and for comparison, it is customary to state the amount of steam consumed per indicated horse-power per hour. By means of the indicator diagram the steam consumption can be computed approximately.

To find the Steam Consumption from the Diagram. The diagram shown in Fig. 40 is from a 20×36 engine, running at a speed of 80 revolutions per minute. A 40-pound spring was used.

By measuring the card, we find the mean ordinate to be .91 inch and the M. E. P. = $.91 \times 40 = 36.4$ pounds.

$$\text{I. H. P.} = \text{Engine Constant} \times \text{M. E. P.} \times \text{piston speed.}$$

$$= .00952 \times 36.4 \times 480.$$

$$= 166.33.$$

In Fig. 40 L M is the atmospheric line and O R the line of zero pressure drawn so that O L = 14.7 pounds. O N is the clearance volume = 8 per cent of the piston displacement. The line P Q is drawn from O R to some point on the compression

curve. From D, a point on the expansion curve before release, the line D F is drawn perpendicular to O R.

Then from the diagram,

$$O R = 3.24 \text{ inches.}$$

$$O F = 3.00 \text{ inches.}$$

$$O P = .345 \text{ inch.}$$

$$D F = .795 \text{ inch.}$$

$$P Q = .795 \text{ inch.}$$

The length of stroke is 36 inches or 3 feet, and the length of the diagram 3 inches. Then 1 inch of the length of the card corresponds to 1 foot of the stroke. The scale of spring used is 40. Therefore we can easily reduce the above dimensions to pounds pressure and to feet.

$$O R = 3.24 \text{ feet.}$$

$$O F = 3.00 \text{ feet.}$$

$$O P = .345 \text{ feet.}$$

$$D F = 31.80 \text{ pounds.}$$

$$P Q = 31.80 \text{ pounds.}$$

The area of the piston (head end) is,

$$\frac{\pi d^2}{4} = \frac{3.1416 \times 400}{4} = 314.16 \text{ sq. in.} = 2.18166 \text{ sq. ft.}$$

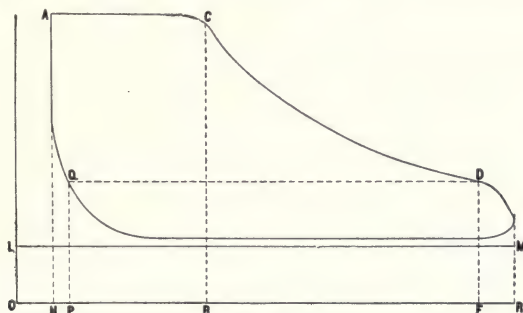


Fig. 40.

We can now find the volume of steam at any point of the stroke.

When the piston is at D, the volume is,

$$2.18166 \times 3 = 6.54498 \text{ cubic feet.}$$

When the piston is at Q, the volume is,

$$2.18166 \times .345 = .75267 \text{ cubic foot.}$$

From the steam tables we can find the weight of a cubic foot of steam at a given pressure.

The weight of 1 cubic foot at 31.8 pounds absolute pressure is .07773 pound. Then the weight of steam present when the piston is at D is,

$$6.54498 \times .07773 = .50887 \text{ pound.}$$

The weight of steam present when the piston is at Q is

$$.75267 \times .07773 = .05852 \text{ pound.}$$

The weight of steam in the cylinder is .50887 pound and the weight of steam kept for compression is .05852 pound. The weight exhausted per stroke is,

$$.50887 - .05852 = .45035 \text{ pound.}$$

The amount used per I. H. P. per hour is,

$$\frac{.45035 \times 2 \times 80 \times 60}{166.33} = 25.993 \text{ pounds.}$$

This may be stated in words as follows :

Measure the pressure from the vacuum line to some point in the expansion curve before release and from the steam tables find the weight of a cubic foot at that pressure. Multiply the volume (in cubic feet) of the cylinder, including clearance to that point, by the weight per cubic foot. The result is the weight of steam in the cylinder. As this weight includes the steam used for compression it must be corrected to obtain the weight used per stroke.

Take some point on the compression curve and measure its absolute pressure. Then compute the weight of steam to this point. Subtract this weight from the weight of steam to the point in the expansion curve and the result is the weight of steam used per stroke.

Multiply the weight of steam used per stroke by the number of strokes and divide by the indicated horse-power as found from the card. The final result is the number of pounds of steam consumed per horse-power per hour.

We may also calculate the steam consumption by taking the point of the expansion curve near the cut-off.

$$O B = 1.23 \text{ inches} = 1.23 \text{ feet of the stroke.}$$

$$B C = 1.8 \text{ inches} = 72 \text{ pounds.}$$

The weight of 1 cubic foot of steam at 72 pounds absolute pressure is .1671 pounds. Then the volume of steam at C is,

$$2.18166 \times 1.23 = 2.68344 \text{ cubic feet.}$$

The weight at C is,

$$2.68344 \times .1671 = .44840 \text{ pound.}$$

The weight kept for compression is the same as previously found, *i. e.*, .05852 pound.

Then the weight of steam used per stroke is,

$$.44840 - .05852 = .38988 \text{ pound.}$$

The steam consumption per I. H. P. per hour is,

$$\frac{.38988 \times 2 \times 80 \times 60}{166.33} = 22.50 \text{ pounds.}$$

If the valve doesn't leak, the amount of steam just after cut off should equal the amount just before release, but our calculation shows that there is $.45035 - .38988 = .06047$ pound more at release than at cut off. This shows that at entrance .06047 pound was condensed before the piston reached C and was re-evaporated before release. This calculation gives an idea of the amount of cylinder condensation.

If there is considerable compression as in Fig. 40 the above method may be simplified by taking the two points D and Q at the same height above the vacuum line. The pressures will then be the same.

Let W = weight of steam used per I. H. P. per hour.

w = weight of one cubic foot of steam at the absolute pressure D F.

L = length of the diagram, N R.

l = distance between Q and D.

P = mean effective pressure.

Then,

$$W = \frac{13,750 \times w \times l}{P L}.$$

Example. A card was taken from a 12×14 engine. Length of card $L = 3.5$ inches, $l = 2.875$ inches. Absolute pressure at

D = 30 pounds. M. E. P. = 30.75 pounds. What is the steam consumption per I. H. P. per hour?

$$\begin{aligned} W &= \frac{13,750 \, w \times l}{P \times L} \\ &= \frac{13.750 \times .0736 \times 2.875}{30.75 \times 3.5} \\ &= 27.03 \text{ pounds.} \end{aligned}$$

This method gives only an approximate steam consumption. On account of the leakage of valves and the initial condensation of steam in the cylinder, the actual consumption is somewhat greater. The excess varies considerably and makes all results so obtained of little value. In our calculation we used one diagram only, that for the head end; we assumed the diagram from the crank end to be the same. The indicated horse-power for each end should be computed, as it is subject to considerable variation. In the above formula the average mean effective pressure should be used. The results of calculations of steam consumption from indicator cards are so unreliable that engineers place little dependence upon them. The results may be used as a basis for estimates, but for accurate knowledge, an engine test must be resorted to.

EXAMPLES FOR PRACTICE.

1. Given the following data to find the M. E. P. Area of card 2.79 square inches, length of card 3.1 inches, scale of spring 50. Ans. 45 pounds.

2. Steam enters a cylinder at a pressure of 210 pounds by gage and leaves at 3 pounds pressure (absolute). What is the thermal efficiency? Ans. 29 per cent.

3. An engine has a B. H. P. of 78.96. The I. H. P. is 86.73. What is the mechanical efficiency. Ans. 91 per cent.

4. An engine develops 149.97 I. H. P. If the piston diameter is $17\frac{3}{4}$ inches, the stroke 30 inches, and the M. E. P. 40 pounds, what is the speed? Ans. 100 revolutions per minute.

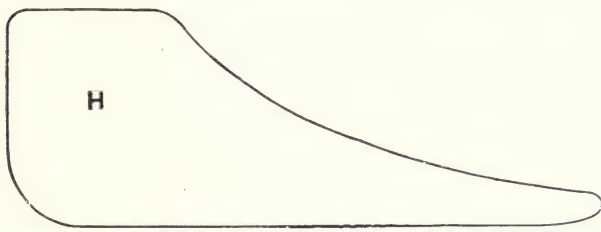
5. The theoretical M. E. P. of a triple expansion marine engine is 48 pounds. What is the probable actual M. E. P.?

6. The cylinders of a triple expansion are 12, 30, and 75 inches in diameter respectively. The stroke is 24 inches. The

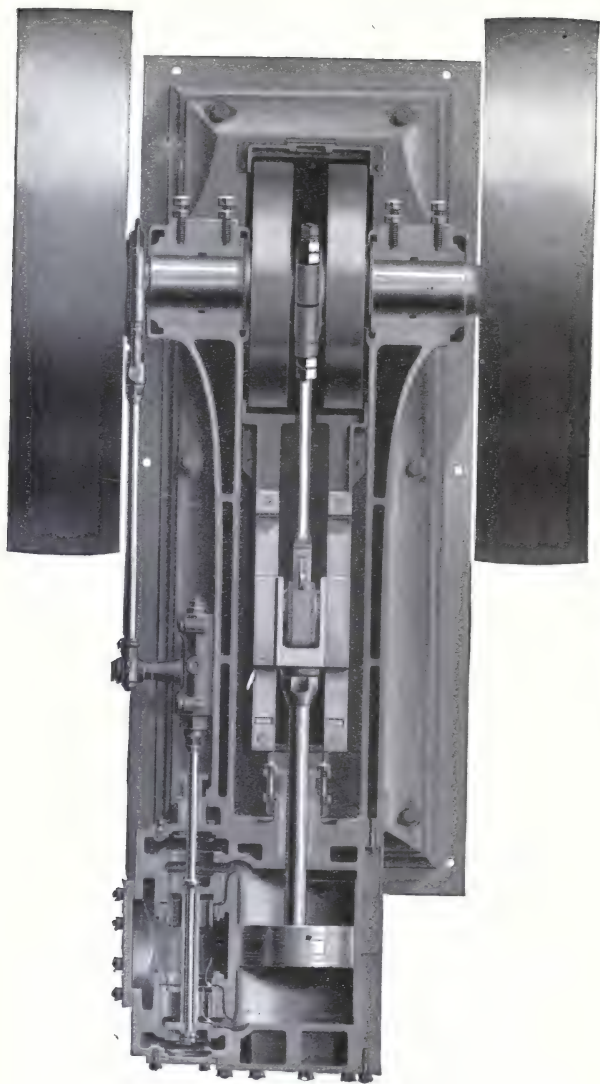
mean effective pressures from the cards were 118.4, 51.6 and 19.34 pounds respectively. What is the I. H. P. when the speed is 125 revolutions per minute? Ans. 2,050 I. H. P.

7. Find the steam consumption from the following card. The engine was running at 75 revolutions per minute, and developing 230 I. H. P. when the card was taken. A 60-pound spring was used, and the M. E. P. for this end was 46 pounds. Assume crank-end card to be the same as the head end.

Ans. 22 pounds (about).



Suggestion: First draw a line .3 inch above the atmospheric line, then reduce this ordinate to *absolute* pressure, and use formula at the bottom of page 55.



HORIZONTAL SECTION OF ENGINE
New York Safety Steam Power Company.

VALVE GEARS

Steam enters the cylinder of the engine through ports which must, in some manner, be opened and closed alternately, in order to admit and exhaust the steam at the proper time. To accomplish this purpose a valve is moved back and forth across the port openings. A complete understanding of the valve and valve gear is essential to the engineer as well as to the designer, for even though a valve be properly designed, its economy may be seriously impaired by improper setting. The design and adjustment of these valves plays a very important part in the efficient action of the steam engine.

The term "valve gear" includes the valve or valves that admit steam to and exhaust it from the cylinder of the engine, together with the mechanism from which the valves derive motion. There may be a single valve to regulate admission and exhaust, or there may be a double set of valves; one set to admit the steam at each end and another to release it. The valve may have a plain reciprocating motion, moved by a rod, or it may be opened by some device that lets go at the proper time, allowing the valve to drop shut under the influence of counter weights, springs or vacuum dashpots. To the first class belong the plain slide valve and its modification of piston valve, gridiron valve, etc.; to the second belong such valves as the Corliss, Brown, and others.

The simplest type of valve is the plain slide or D valve as shown in Fig. 1.

In this figure V is the valve, R the valve rod, K the exhaust cavity, P and P' the steam ports, E the exhaust port, AB the valve seat, and DM the bridges of the valve seat. The valve seat must be planed perfectly smooth, so that pressure on the valve will

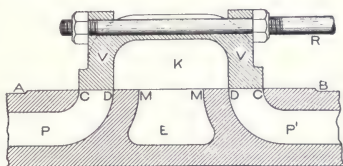


Fig. 1.

make a steam tight fit, and cause as little friction as possible when the valve slides. Furthermore, the length of the seat AB must be a little less than the distance from the extreme right-hand position of the right-hand edge of the valve to the extreme left-hand position of the left-hand edge of the valve. This allows the valve at each stroke slightly to over travel the seat, thus keeping it always worn perfectly flat and smooth. If the valve seat were not raised slightly above the rest of the casting, or if it were too short, the constant motion of the valve would soon wear a hollow path in the valve seat, and it would cease to be steam tight.

Eccentric. The valve usually receives its motion from an eccentric which is simply a disc, keyed to the shaft in such a

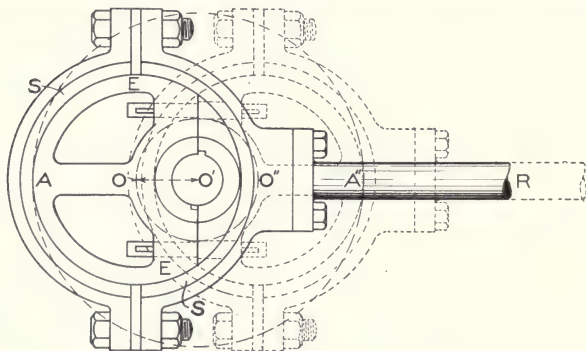


Fig. 2.

manner that the center of the disc and the center of the shaft do not coincide. It is evident that as the shaft revolves, the center of this eccentric disc moves in a circle about the shaft as a center, just as if it were at the end of a crank. The action of the eccentric is equivalent to the action of a crank the length of which is equal to the eccentricity of the eccentric (the distance between the center of the eccentric and that of the shaft).

Fig. 2 represents the essentials of an ordinary eccentric. O is the center of the shaft, O the center of the eccentric disc E, and S is a collar encircling the eccentric and attached to the valve rod R.

As the eccentric turns in the strap, the point O moves in the dotted circle around O', and the point A also moves in a circle. When half a revolution is accomplished the point O will be at O'', the point A will be at A'', and the eccentric strap and valve rod will be in the position indicated by the dotted lines.

Since the revolving shaft transmits motion to the valve through the eccentric, it will be necessary to study the relative motions of the crank and eccentric in order to get a clear idea of the steam distribution.

The distance of the center of the eccentric from the center of the shaft (OO' in Fig. 2) is known as the eccentricity, or throw, of the eccentric. The travel of the valve is twice the eccentricity.

Valve without Lap. Fig. 3 shows a section through the steam and exhaust ports of an engine, together with a plain slide

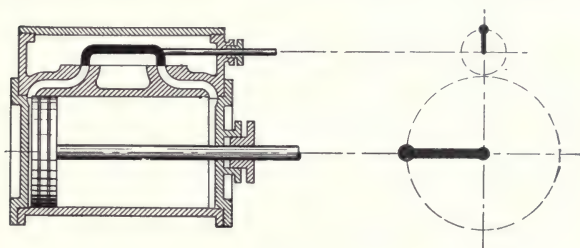


Fig. 3.

valve placed in mid-position, and so constructed that in this position it just covers the steam ports and no more. A valve is in mid-position when the center line of the valve coincides with the center line of the exhaust port.

Fig. 1 shows the same valve drawn to a larger scale.

Suppose the valve is moved a slight distance to the right; the port P (see Fig. 1) is then uncovered and opened to the live steam which enters the cylinder and causes the piston to move. Since the two faces of the valve are just sufficient to cover the steam ports, it is evident that as the port P opens to live steam, the port P' opens to the exhaust. The ports are closed only when the valve is in mid-position. This allows admission and exhaust

to continue during the whole stroke. With such a valve there is no expansion or compression; the indicator card would be a rectangle, and the M. E. P. would be equal to the initial steam pressure, assuming no frictional losses in the steam pipe or condensation in the cylinder.

For a theoretical discussion of valve motion, it is assumed that the eccentric rod moves back and forth in a line parallel to the center line of the engine. This is not the case in practice, for the eccentric rod always makes a small angle with the center line, just as the connecting rod does, but the eccentricity is so small in comparison with the length of the eccentric rod that the angularity of the eccentric rod is very much smaller than the angularity of the connecting rod, and its influence may be neglected without appreciable error.

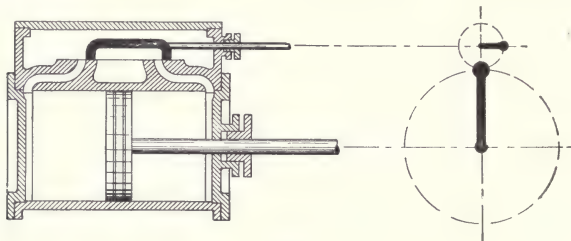


Fig. 4.

When the valve shown in Fig. 3 is in mid-position, the crank is on dead center, the eccentric is set at right angles to it, and the piston is just ready to begin the stroke.

Fig. 4 shows the relative positions of crank, piston, eccentric and valve when the crank has made a quarter turn or the piston has moved to half stroke. The eccentric is now in its extreme position to the right, the valve has its maximum displacement and both the steam and exhaust ports are wide open. The valve will not close again until the piston has reached the end of its stroke.

This type of valve is used only on small and unimportant engines, and since it allows no expansion of the steam, is very uneconomical. Furthermore, it will be seen that this valve opens just after the stroke begins, which is impractical, for it means that the piston has begun its stroke before the full steam pressure

reaches it, which will cause an inclined admission line on the indicator diagram.

Valve with Lap. If the face of the valve is made longer than shown in Fig. 1, so that in mid-position it overlaps the steam ports, we shall have a valve such as shown in Fig. 5. The amount that the valve overlaps the steam ports is called the lap of the valve. In Fig. 5, DI is the *inside* lap, and OC the *outside* lap. It will at once be seen that both the admission and exhaust ports may remain closed during a part of the stroke, thus making expansion

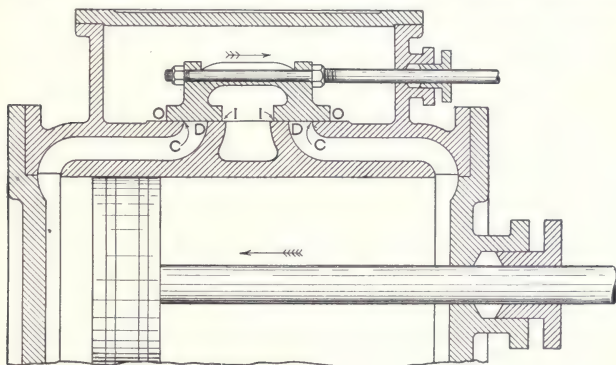


Fig. 5.

and compression possible. It is also evident that steam cannot be admitted until the valve uncovers the port by moving a distance from mid-position equal to OC. Admission continues until the valve returns to such a position that the outer edge of the valve again closes the port. Release will begin when the inner edge of the inside lap begins to uncover the port.

Fig. 6 represents a valve with lap, at the point of admission. Since the valve must move a distance equal to the outside lap before admission can take place, it is evident that the eccentric can no longer be at right angles to the crank at the beginning of the stroke, but must be ahead of the right-angle point by an amount equal to AOC. The angle AOC is known as the angular advance.

The maximum displacement of the valve is attained when the eccentric is horizontal as shown in Fig. 7. In this position

both the steam and exhaust ports are wide open, and any further motion of the piston will cause the valve to move toward its mid-position.

Admission continues until the valve returns to the position

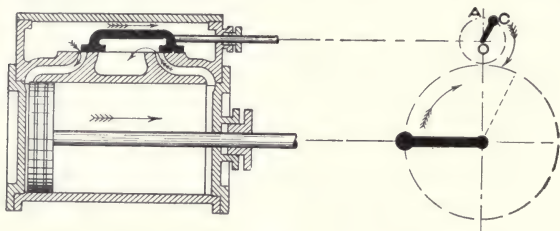


Fig. 6.

shown in Fig. 8. Here the outside lap just closes the left-hand steam port, cut-off takes place, and the steam already in the cylinder begins to expand. As the valve continues to move toward the left, the left-hand inside lap begins to uncover the left-hand port and releases the steam at the position shown in Fig. 10.

The dotted lines of Fig. 7 show the valve in its extreme

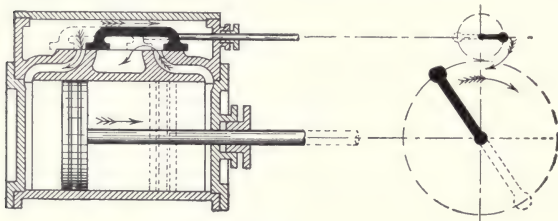


Fig. 7.

position to the left. Any further movement of the piston will cause it to return toward mid-position.

The dotted position of crank and eccentric in Fig. 10 shows the valve returned to the point of compression, which continues until the conditions of Fig. 6 are again reached and the opening valve allows steam again to enter the cylinder.

This process has been traced step by step for one end only; let us now consider what is happening at the other end.

Admission is the point at which the valve opens to admit steam to the cylinder. **Cut-off** is the point at which the valve closes to cut off the admission of steam. **Release** is the point at which the exhaust is opened; and **Compression** is the point at which the exhaust is closed.

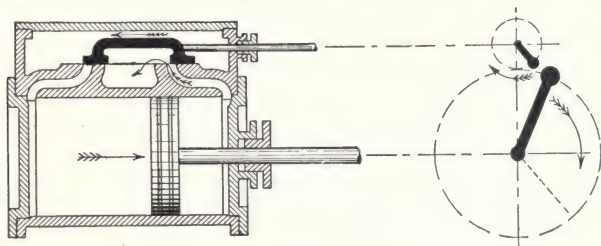


Fig. 8.

While the crank is moving from the position shown in Fig. 6 to that of Fig. 8, steam is being admitted to the head end and being exhausted from the crank end. The inside lap being less than the outside lap, causes the exhaust to continue longer than the admission.

Fig. 9 shows the relative positions of crank, eccentric and valve when the exhaust closes on the crank end and compression

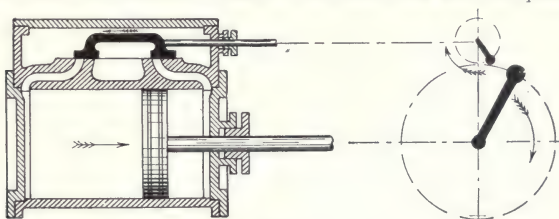


Fig. 9.

begins. Between these two positions the steam is expanding in the head end and exhausting from the crank end.

Between the positions of Fig. 9 and Fig. 10 both ports are entirely closed, expansion is taking place in the head end and compression in the crank end. Fig. 10 is head-end release. Fig. 11 shows admission at crank end of cylinder and marks the end of crank-end compression.

By referring to Figs. 6-11, the effect of any change of lap may at once be observed. If the outside lap is *increased*, the valve must move farther from mid-position before admission will occur, and on the return, after the maximum displacement is reached, the outside lap, being wider, will close the port sooner, and the cut-off shown in Fig. 8 will take place before the crank

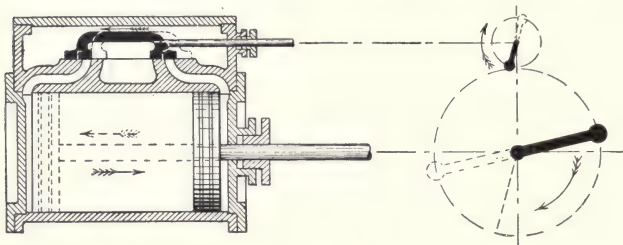


Fig. 10.

reaches the angle there shown. A decrease of outside lap will make cut-off later and admission earlier.

If the inside lap is increased, the valve must move farther before release occurs and the crank angle would be greater than shown in Fig. 10. On the return of the valve to the dotted position shown in Fig. 10, the port will close earlier and make an earlier compression; the crank angle will be less than is there shown. Decreasing inside lap will cause earlier release and later compression.

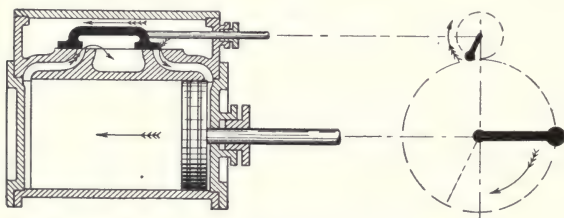
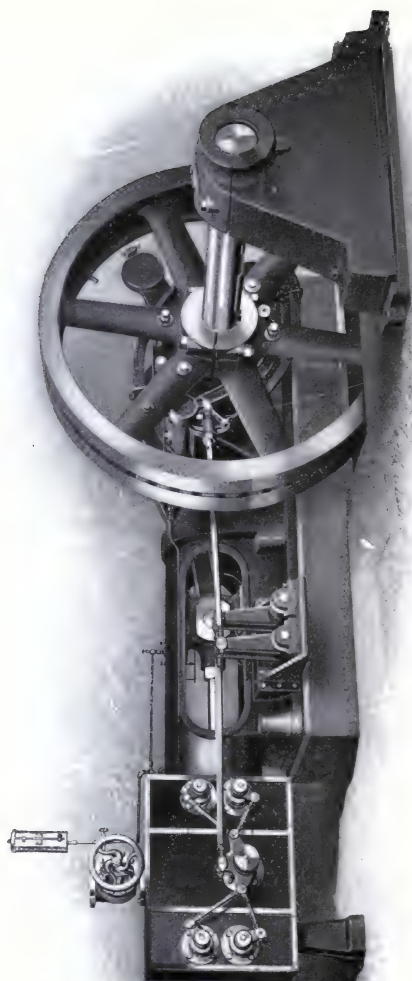


Fig. 11.

Thus we see that it is the outside lap that influences admission and cut-off, and the inside lap that controls release and compression. For this reason the outside lap is often called the *steam lap*, and the inside lap the *exhaust lap*.



SIMPLE CORLISS VALVE ENGINE.
Ball and Wood Co.

Lead. If a valve having lap is in mid-position, the port is closed and the engine cannot start because no steam can enter the cylinder. That the steam may be ready to enter the cylinder at the beginning of the stroke it is necessary that the eccentric be set more than 90° ahead of the crank and the eccentric radius will take an angle as shown in Fig. 6, called the *angular advance*. In order that the ports and clearance may be properly filled with steam at the beginning of the stroke, it is necessary that the valve be displaced from its mid-position an amount slightly greater than the outside lap. With the piston at the end of the stroke the valve will have a position as shown in Fig. 12. The port will be open the distance AB. This causes the eccentric to be moved forward a slight amount in excess of the angular advance. This excess is called the *angle of lead*.

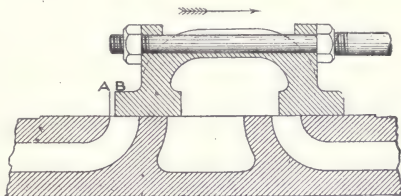


Fig. 12.

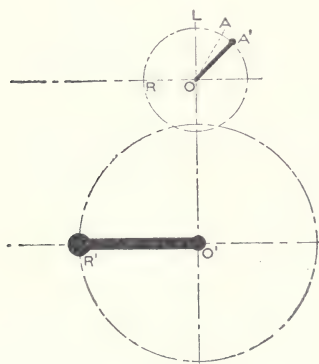


Fig. 13.

In Fig. 13, $O'R'$ represents the crank at the beginning of the stroke, LOA the angular advance, and AOA' the angle of lead. The eccentric, to give lead, must be set at the angle ROA' ahead of the crank or 90° plus angular advance plus angle of lead. In large, quick-running engines, a liberal lead is essential, so that the ports and clearance may be well filled with steam before the stroke begins. If there is no lead, a portion of the steam will be used in filling these places and full

pressure steam will not reach the piston until it is well advanced on the stroke. This will give a sloping admission line as shown

in Fig. 14. Too much lead, on the other hand, will cause too early an admission as shown in Fig 15.

If the angular advance is increased, the eccentric will be moved farther ahead of the crank, and consequently will begin its motion sooner. It will necessarily arrive at each of the events



Fig. 14.

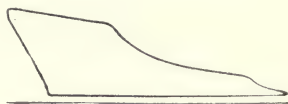


Fig. 15.

sooner than before. If then, the angular advance is increased, all of the events of the stroke will occur earlier.

Inequality of Steam Distribution. In the valve diagrams thus far considered, the events of the stroke have been discussed for each end separately, without reference to the relation of similar events on the other side of the piston.

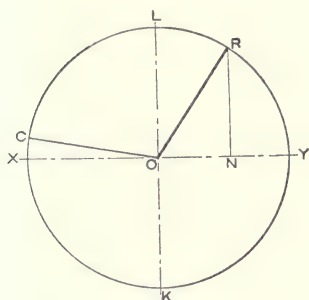


Fig. 16.

If the connecting rod were of infinite length, so that it would always remain parallel to the center line of the engine, the distribution would be the same for both ends of the cylinder. In practice, the connecting rod is from 4 to 8 times the length of the crank, which causes the connecting rod always to be at an angle to the center line of the engine, and for a given crank angle makes the piston displacement greater at the head end than at the crank end.

To find the displacement of the valve, let us consider Fig. 16. The circle represents the path of the eccentric center during a complete revolution of the engine. OC represents the crank, and OR the corresponding position of the eccentric. The diameter XY represents the extent of the valve travel. Since the eccentric rod is so long in comparison to the eccentricity, we make no appreciable error by assuming it always to be parallel to the center

line of the engine. When the eccentric is at OL , the valve is in mid-position. At OR the valve has moved from mid-position an amount ON , found by dropping a perpendicular from R to the center line XY . If the angularity of the connecting rod could be neglected, the piston displacement could be found in the same manner.

To find the displacement of the piston, a diagram as shown in Fig. 17 must be drawn. In this figure AB represents the cylinder, P the piston, II the crosshead, HR the connecting rod, and OR the crank. Suppose now the engine should stop in this position and then be clamped. The piston displacement would be represented by AP . If the crank pin at R should now be loosened so as to allow the connecting rod to fall to a horizontal position, the point R would describe the arc of a circle RN , and XN would represent the piston displacement and would be equal to AP .

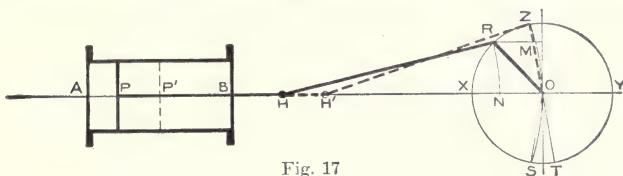


Fig. 17

Suppose now that in this disconnected way the piston, crosshead and connecting rod were moved forward until the end of the rod came to O . P would then be at P' and the piston would be in the middle of its stroke. Now swing the end of the rod up to its proper position on the crank-pin circle, the piston remaining stationary. It would describe an arc OZ . The crank pin would be at Z , less than a quarter revolution from X , while the piston would be in the middle of its stroke.

Suppose this engine were running with cut-off at half stroke on the head end and that XOZ represented the corresponding crank angle. On the return stroke the valve would cut off at the same crank angle $YOT = XOZ$, and OT would represent the crank cut-off on the return or crank-end stroke. The piston, as we have just seen, will not be at half stroke except when the crank is at OZ or OS . Consequently OT is less than half stroke and cut-off takes place earlier at the crank end than at the head end. When the crank is at OZ the eccentric will be at OA (Fig 17*a*), and the valve displace-

ment will be OB . When the crank is at OT the eccentric will be at OA' , and the valve displacement will be OB' , which is equal to OB , the displacement of the valve at cut-off in the head end. The piston displacement will be OX in the head end and WY in the crank end when cut-off occurs. If the connecting rod always remained parallel to the center line, the cut-off would be the same at both ends.

Compensation of Cut-off. It has already been pointed out that lengthening the outside lap makes the cut-off earlier, and shortening the lap makes it later. The cut-off in the case just cited may then be equalized by altering the outside laps. If we increase the outside lap on the head end, or decrease the crank-end lap, the inequality will be less. By changing either or both of the laps the proper amount, the cut-off may be exactly equalized.

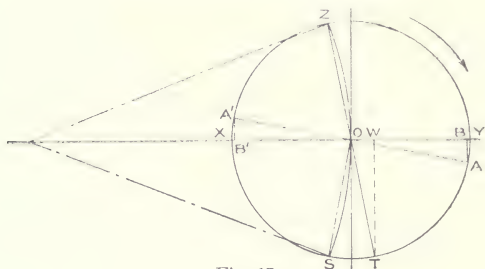


Fig. 17a.

But altering the outside lap changes the lead as has already been explained. If the lap is increased on the head end, the lead will be less than on the crank end. If the lead becomes too small on the head end, the angular advance may be increased but the inequality of lead will still remain, for this increase of angular advance will increase the lead at the crank end as well as at the head end, and by hastening all the events of the stroke may give a bad steam distribution if care is not taken.

Unequal lead is of less consequence on a low-speed than on a high-speed engine. On low-speed engines the cut-off may be equalized at the expense of lead with beneficial results, but on high-speed engines it will not do to give too little lead at one end. A high-speed engine requires more lead than a low-speed, for there is relatively less time in each stroke for the clearance to fill with steam.

If both inside laps are equal, compression will not occur equally at both ends. To equalize it, the inside laps may be changed in the same manner as the outside laps are changed to equalize the cut-off. By altering these inside laps to equalize compression, it may happen that the lap is reduced enough to leave the exhaust port open when the valve is in mid-position.

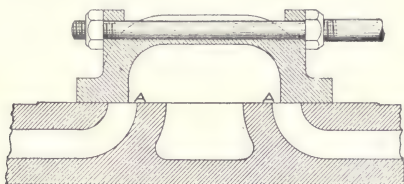


Fig. 18.

This opening of the valve is called an inside clearance, or *negative lap*. In Fig. 18, A is the inside clearance.

Rocker. Sometimes it happens that the valve stem and eccentric rod cannot be so placed that they will be in the same straight line; or it may be that the travel of the valve must be so great as to require an excessively large eccentric. In such cases a rocker may be used.

Fig. 19 shows a valve that is not in line with the eccentric. This occurs in horizontal engines when the valve is set on top of

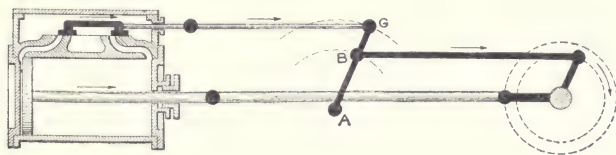


Fig. 19.

the cylinder instead of on one side. By means of the rocker AG the valve may receive its proper motion.

In case it is more convenient to place the pivot of the rocker arm between the connections to the valve stem and those of the eccentric rod, such an arrangement as shown in Fig. 20 may be used. Here it will be noticed that the valve stem and eccentric rod are moving in opposite directions, and to give the valve the

same motion as in Fig. 19, the eccentric must be moved 180° ahead of the position there shown.

If AB is less than AG , the valve travel will be greater than twice the eccentricity, in proportion as AG is greater than AB . In all cases the valve travel is to twice the eccentricity as AG is to AB . Thus, if the valve travel is $4\frac{1}{2}$ inches, AB , 15 inches, and AG , 18 inches, then $\frac{15}{18} \times 4\frac{1}{2} = 3\frac{3}{4}$ inches, will equal twice the eccentricity.

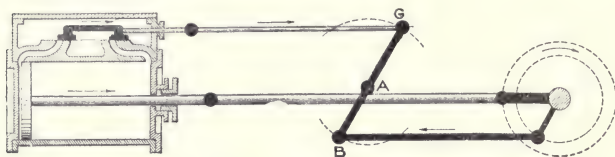


Fig. 20.

A valve gear may be so laid out as to make both the cut-off and the lead equal for both ends of the cylinder. This may be done by a proper proportion between the rocker arms, and a careful location of the pivot of the rocker. The eccentric must then be set accordingly. In this manner the Straight Line engine equalizes the cut-off and lead. A discussion of this method will be considered later.

VALVE DIAGRAMS.

Zeuner's Diagram. In order to study the movements of valves, the effect of lap, lead, eccentricity, etc., diagrams of various sorts have been devised. By the use of diagrams we may acquire a knowledge of valve motion without the complex mathematical expressions that such a discussion would entail. The most useful of these various diagrams is that devised by Zeuner, and to avoid complexity we shall confine ourselves to a discussion of this diagram alone. The eccentric rod is assumed to be of infinite length, and the positions of the crank are shown on the diagrams. The displacement of the piston can easily be found if the ratio of crank to connecting rod is known.

In Fig. 21 let OY be the eccentricity, then XOY will represent the valve travel, and the center of the eccentric will move

in the circle XWY. Let OR represent the position of the crank and Or the corresponding position of the eccentric, which is $90^\circ +$ angle of advance θ ahead of the crank. Draw OW perpendicular to XY and lay off from it the angle WOM = angle of advance θ towards the crank. With OM as a diameter, construct a circle. OM is equal to the eccentricity, and the circle MPO is known as the *valve circle*. If OR, the center line of

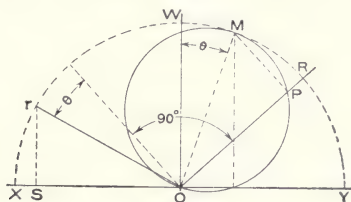


Fig. 21.

the crank, cuts this valve circle at P, then OP is equal to the displacement of the valve from mid-position.

To prove this, draw OS perpendicular to XY. Since Or is the position of the eccentric, OS will represent the valve displacement from mid-position. Draw MP. Then by geometry OPM is a right angle because it is inscribed in a semicircle. OSr is also a right angle; the two right-angled triangles OSr and OMP are equal because they are similar and have two corresponding sides equal. Or = OM, being radii of the same circle. But we have seen that OS is equal to the valve displacement, therefore OP is also equal to the valve displacement, for it is equal to OS.

Now that the truth of our proposition has been proved, let us see how we may study the valve motion from such a diagram. See Fig. 22. As before, let XY represent the valve travel, then the circle XEYF will represent the path of the center of the eccentric. Let θ be the angular advance and lay off EO toward the crank, making an angle θ with the vertical. Produce EO to F, and on OE and OF as diameters draw the valve circles as shown. Let the outside lap be an amount equal to OV, then with O as a center and OV as a radius draw an arc intersecting the upper valve circle at V and K. Lay off OP equal to the inside lap and

with O as center and OP as a radius draw an arc intersecting the valve circle at P and Q . Draw the crank line AO passing through V . Then, when the crank is in this position, the displacement of the valve is equal to OV (the outside lap) and the steam is ready to enter the cylinder. This is the position of the crank at admission, and the crank angle XOA is called the lead angle. The valve has lead, therefore the admission takes place before the end of the stroke. When the crank reaches the position OE , the displacement of the valve is equal to OE , the eccentricity, and is

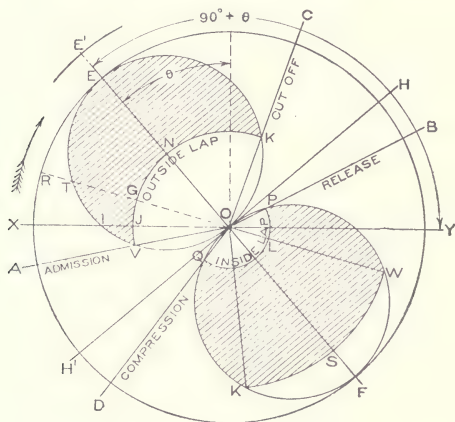


Fig. 22.

the maximum displacement. Further motion of the piston causes the valve to move toward mid-position until, at the crank position OC , the displacement OK is again equal to the outside lap and the valve has reached the point of cut-off. When the position OH is reached, the crank line is tangent to both valve circles and there is no displacement of the valve. At this point the valve is in mid-position.

Further crank movement draws the inside lap toward the edge of the exhaust port until, at the crank position OB , the displacement is equal to OP (the inside lap) and release begins. At OF the maximum valve displacement is again reached and the valve moves in the opposite direction until at OD its displacement

from mid-position, is again equal to $OQ = OP =$ the inside lap, and compression takes place. At OII' the valve is again in mid-position. At OX the displacement of the valve is OI , but since the valve has to move the distance OJ before the port begins to open IJ must represent the port opening when the crank is on dead center and by definition we know that lead is the amount of port opening at this position. Therefore IJ represents the lead.

At the position R, the port is open an amount equal to TG, at E the opening is a maximum equal to EN. At C the opening is nothing. If LW represents the total width of the steam port, the exhaust port will be open wide when the displacement of the valve is equal to OW and it will remain wide open while the crank swings from OW to OK.

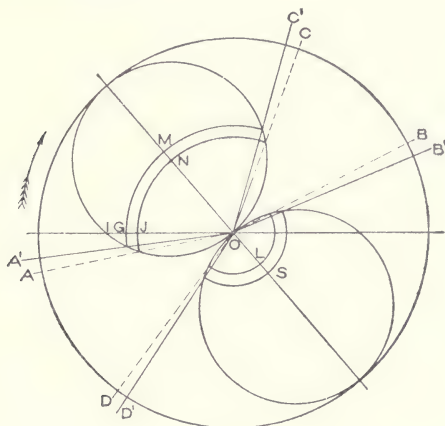


Fig. 23.

If the width of steam port in addition to the outside lap were laid off on the other valve circle it would fall at E' . For the admission port to be wide open, the displacement of the valve would have to be equal to OE' , which is more than the maximum displacement. This shows that in this case the steam port is never fully open and that the left-hand edge of the valve overlaps the right-hand edge of the port by an amount equal to EE' when the valve has reached its maximum displacement.

Fig. 22, with its two valve circles, shows the diagram for the head end of the cylinder only. The crank-end diagram would be similar except that the laps might not be equal to those of the head end.

We are now in a position to consider more in detail the effect of changing in any way either the valve or the setting. Let us consider Fig. 23, which is in every way like Fig. 22 except that all unnecessary letters and lines are omitted to avoid confusion. If the outside lap is increased an amount equal to NM, the admission will take place later, at crank position OA' ; the lead will

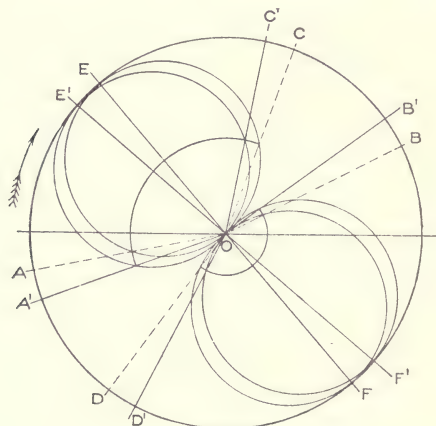


Fig. 24.

be reduced to IG and cut-off will take place earlier at OC' . If the outside lap is *reduced* a like amount the contrary effects will be observed. If the inside lap is increased an amount equal to LS, the release will take place later at the crank position OB' and compression will take place earlier at OD' . The contrary effect will be observed by decreasing the inside lap.

If the angular advance is increased (see Fig. 24) all the events will occur earlier. This is evident from the figure; the crank revolves in the direction indicated by the arrow and OA' (new position of admission) is *ahead* of OA , the old position.

If the eccentricity is increased, Fig. 25, the valve travel will increase and admission will take place earlier at OA' ; the lead will be increased an amount equal to II' , and cut-off will take place later at OC' . Release will be earlier at OB' and com-

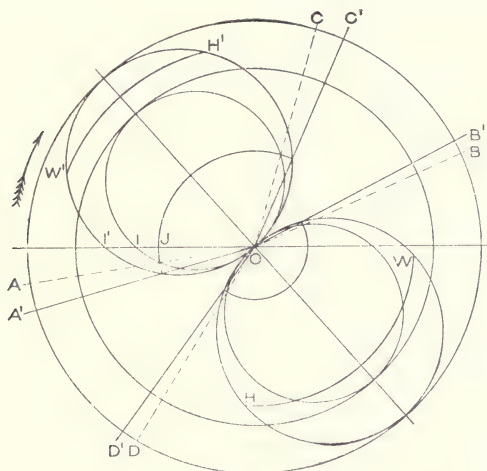


Fig. 25.

pression will be later at OD' . The upper valve circle will now cut the arc drawn from O as a center, with a radius equal to the outside lap plus the width of steam port, in the points W' and H' , and the admission port will be open wide while the crank is moving from OW' to OH' . Similarly, the lower valve circle cuts the arc drawn from O as a center, with a radius equal to the inside lap plus the width of steam port, in the points W and H. The steam port is then wide open to exhaust while the crank is moving from W to H. From the above it will be seen that the periods are all changed by changing the travel; thus, admission and exhaust begin sooner and last longer, while expansion and compression begin later and cease sooner. With change in the angular advance, however (see Fig. 24), the periods are neither increased nor decreased.

For convenience, these results are collected in the following table which shows the effect of changing the laps, travel, and angular advance:

	Increasing Outside Lap.	Increasing Inside Lap.	Increasing Travel.	Increasing Angular Advance.
Admission.	Is later. Ceases sooner.	Not changed.	Begins earlier. Continues longer.	Begins earlier. Same period.
Expansion.	Is earlier. Continues longer.	Beginning unchanged. Continues longer.	Begins later. Ceases sooner.	Begins earlier. Same period.
Exhaust.	Unchanged.	Occurs later. Ceases sooner.	Begins earlier. Ceases later.	Begins earlier. Same period.
Compression	Begins at same point.	Begins sooner. Continues longer.	Begins later. Ceases sooner.	Begins earlier. Same period.

PROBLEMS.

All the problems on valve gears involve the relations between certain variables which are :

The valve travel.

Angle of lead.

Outside lap.

Inside lap.

Points of stroke at which admission cut-off, release and compression take place.

In designing a Slide Valve, a few of these variables depend upon the conditions under which the engine is to run. For instance, the valve travel is limited, cut-off must be at a certain point and the engine must have a certain lead. Then, with the aid of a Zeuner's diagram, the remaining proportions of the valve may be determined.

Let us consider a few examples:

Given the valve travel = 3 inches.

Inside lap = $\frac{3}{4}$ inch.

Angular advance = 35°

Angle at cut-off = 115°

To determine the laps, the lead and the crank angles at admission, compression and release.

In Fig. 26, let XY represent the valve travel = 3 inches. Draw OM perpendicular to XY , and on XY as a diameter draw the circle $XMYF$ representing the path of the center of the eccentric as it revolves about the shaft. Lay off the angle $MOE =$ the angular advance = 35° so that the angle XOE is equal to 90°

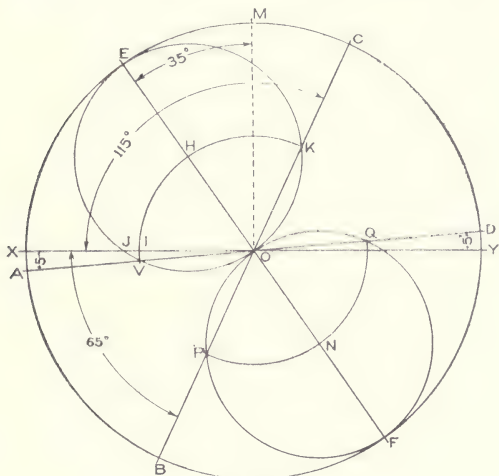


Fig. 26.

minus the angular advance. Produce EO to F . Then on OE and OF as diameters draw the valve circles. The eccentricity OE or OF , if no rocker is used, will be half the valve travel. Lay off the crank angle $XOC =$ angle of crank at cut-off = 115° , and OK will then represent the distance of the valve from mid-position when cut-off takes place. This distance we know is the outside lap. Draw the arc KI , known as the lap circle, and it will cut the valve circle again at V . When the valve is again the distance $OV =$ the outside lap from mid-position, admission will take place. Draw the line OVA and this will represent the position of the crank at admission.

When the crank is at OX, the valve displacement is equal to OJ. This is at dead center and the valve is open the amount IJ, for it has moved this distance more than the outside lap. Therefore IJ is the lead for this end.

Now on the other valve circle, draw the arc PQ with the inside lap ($\frac{3}{4}$ inch) as a radius. It will cut the valve circle at P and Q. When the valve displacement is equal to OQ, the exhaust port has just closed and the engine is at compression. In the same way OP is the valve displacement at release when the port begins to open. OQD represents the crank position at compression and OPB the crank position at release.

The results then are as follows :

Given :

Valve travel	= XY = 3 inches.
Angular advance	= angle MOE = 35° .
Inside lap	= OP = $\frac{3}{4}$ inch.
Crank angle at cut-off	= angle XOC 115°

Found :

Outside lap	= OK = $\frac{3}{4}$ inch.
Angle of lead	= XOA = 5° .
Linear lead	= IJ = $\frac{3}{32}$ inch.
Max. port opening for admission	= HE = $\frac{3}{4}$ inch.
Crank angle at compression	= XOD = 185°
Crank angle at release	= XOB = 65°
Max. port opening for exhaust	= FN = $\frac{3}{4}$ inch.

Fig. 26 is drawn full size, and all of these measurements may readily be verified. This figure is drawn for the head end only. If the crank angle at cut-off is the same on both ends, the Zeuner's diagram for the crank end will be exactly like Fig. 26.

ANOTHER PROBLEM.

Given :

The valve travel	= 3 inches.
The lead angle	= 6° .
Crank angle at cut-off	= 70° .
Crank angle at compression	= 75° .

To Find :

Angular advance.
Laps.
Linear lead.
Crank angle at release.

As before, let XY represent the valve travel = 3 inches and draw OM and the circle XMYF. See Fig. 27. Lay off the lead

angle $XOA = 6^\circ$. Then OA represents the crank position at admission. Next lay off the crank angle XOC , the angle at cut-off 70° . Bisect the angle COA by the line OE and on OE draw the valve circle. Angle $MOE =$ the angular advance. The valve circle will cut the crank lines OC and OA at K and V respectively. If the work has been carefully done, OK will be exactly equal to OV and will represent the outside lap. The lead is IJ as before. Draw OD at position of the crank at compression so that angle $XOD = 75^\circ$. Continue OE to cut the eccentric circle at F . On

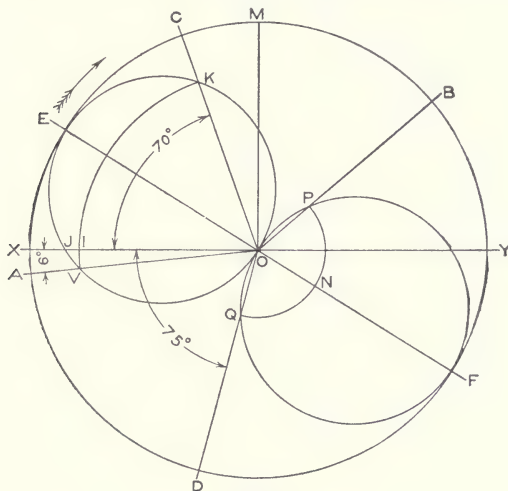


Fig. 27.

OF draw the second valve circle. It will cut OD at Q , and OQ will represent the inside lap. Draw the lap circle OP , and the crank position OPB . This will be the crank position at release.

The angular advance in this problem is large and all the events of the stroke are early. Compression and release are excessively early and the outside lap is unusually large. In the previous problem, with cut-off at about two-thirds stroke, the results were nearly normal. Cut-off with the plain slide valve, earlier than half stroke cannot be had without sacrificing the steam distribution on the other events.

To sum up we have

Given :

Valve travel	= XY	= 3 inches.
Lead angle	= XOA	= 6°.
Crank angle at cut-off	= XOC	= 70°.
Crank angle at compression	= XOD	= 75°.

To find :

Angular advance	= MOE	= 58°.
Outside lap	= OV	= 1 $\frac{3}{8}$ inches.
Lead	= IJ	= $\frac{3}{8}$ inch.
Inside lap	= OQ	= $\frac{3}{8}$ inch.
Crank angle at release	= XOB	= 130°.

Suppose in this last problem the cut-off had been given at half stroke instead of having the crank angle given, and that the compression had been given in the same way. We should, of course, need to know the ratio of length of connecting rod to crank. Let this be given as 4, that is, the connecting rod is four times the length of the crank.

In Fig. 28 let XY represent the valve travel. Extend XY to the left to the point Z, and make OZ equal to four times OX. With Z as a center and OZ as a radius, strike an arc OC that will cut the eccentric circle at C; then draw OC, which will represent the crank when the piston is at half-stroke, which is assumed to be the point of cut-off.

To find the crank angle at compression, lay off YH equal to .8 of the distance YX. From H lay off HW = OZ = four times OX. From W as a center with a radius WH, draw an arc cutting the eccentric circle at D. Draw OD, which will represent the position of the crank at compression.

The student is advised to read over again pages 13 to 14 if this explanation of finding the crank angle does not seem perfectly clear.

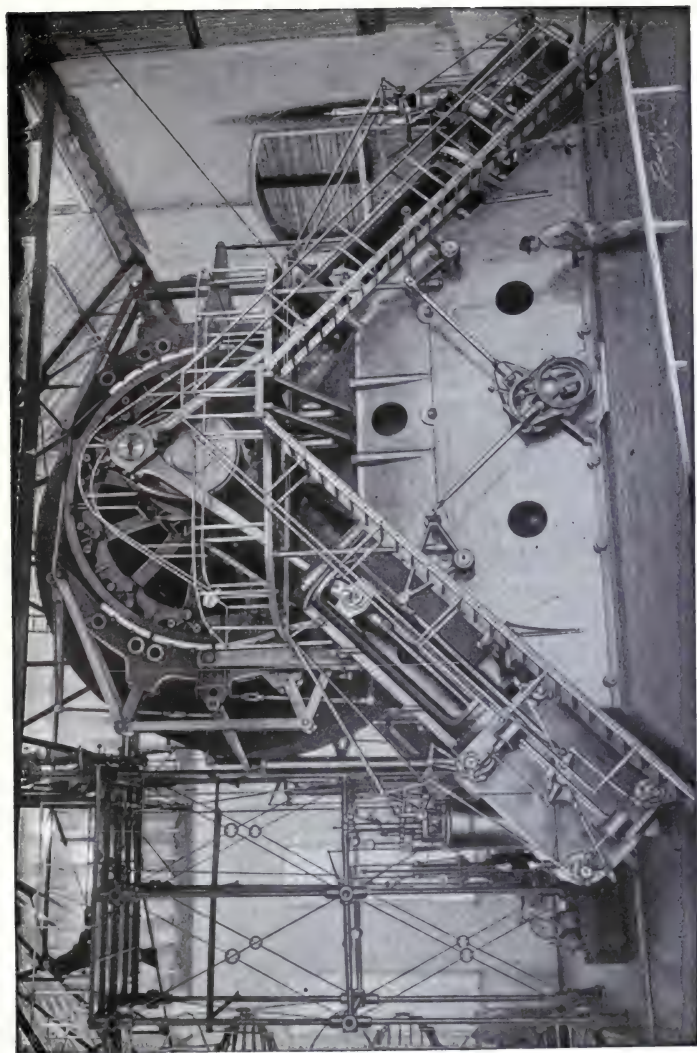
ANOTHER PROBLEM.

Given :

Cut off at .6 stroke.	
Lead	= $\frac{1}{16}$ inch
Maximum port opening	= $\frac{1}{2}$ inch.
Ratio of crank to connecting rod	= 4.

To find :

The eccentricity.
Lead angle.
Angular advance.
Laps.



TAMARAK HOISTING ENGINE AT MINE.
Nordberg Manufacturing Co

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Substituting the figures we have $.5 : .75 :: x : 2\frac{1}{8} \therefore x =$ the probable eccentricity; equals 1.42 inches.

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OC at K', and OK' will be the new outside lap and I'J' will be the new lead (assuming the lead angle to be 7°). This lead I'J' is $\frac{1}{8}$ inch, while the required lead is only $\frac{1}{10}$ inch. Now decrease the angular advance enough to correct one half of this difference, by drawing a new lap circle J''K'' of $\frac{1}{10}$ inch greater radius. This will make the valve circle cut OC at K'', so that OK'' will now be

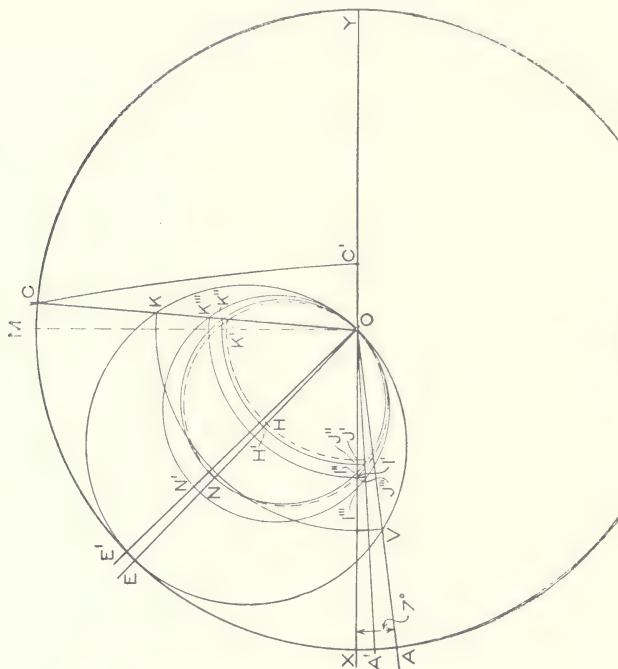


Fig. 29.

the final lap, and I''J'' the final lead, which is equal to the required $\frac{1}{10}$ inch. The lead angle is now XOA' instead of XOA. The port opening at N II' is $\frac{1}{2}$ inch (nearly) as required, but the change in angular advance necessitates an increase of lap if cut-off is to remain the same. This reduces the port opening by the amount III', so that the maximum opening is only .46 inch. By increasing the eccentricity this port opening may be increased.

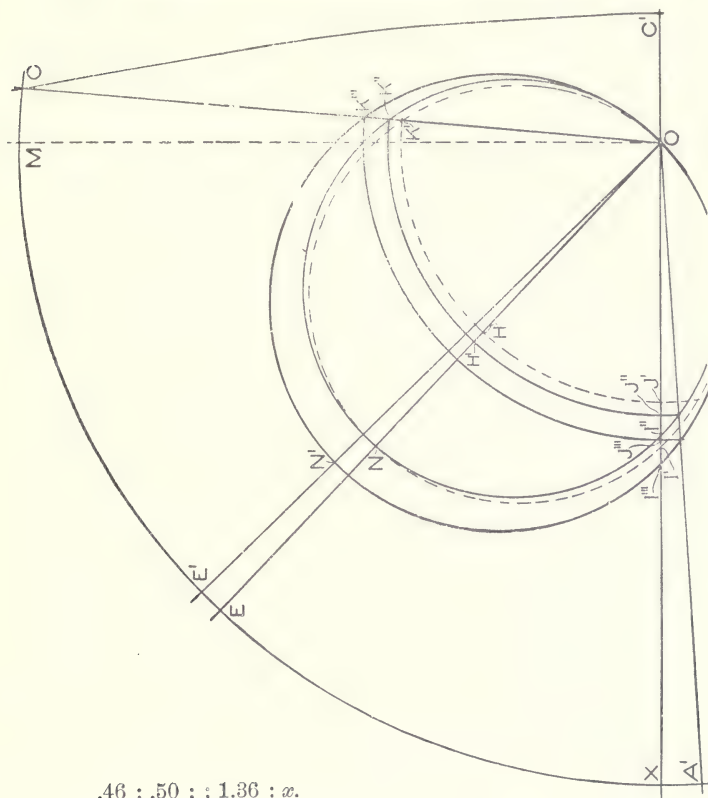


Fig 29c. Essentials of Fig. 29 to larger scale.

$$.46 : .50 :: 1.36 : x.$$

$x = 1.48$, the true eccentricity.

Now draw the valve circle on OE' with a diameter of 1.48 inches. It will cut OC in K''' and OX in J''' . The lap will be $OK''' = .97$ inch, the lead will be $I''J''' = \frac{1}{16}$ inch, the angular advance will be MOE' and the eccentricity ON' .

To sum up we have

Given:

The cut-off = .6 stroke.

The lead = $\frac{1}{16}$ inch

Max. port opening = $1\frac{1}{2}$ inch.

Obtained all of the above conditions together with:

Lap	= .97 inch.
Lead angle	= $\angle XO A'$
Angular advance	= $\angle MO E'$

Compression, release and inside laps are found as in the previous problems.

There are of course all sorts of combinations that would make up different problems, but they can all be solved in the same general way, as they are modifications of the above.

DESIGN OF THE SLIDE VALVE.

In designing a slide valve some of the variables are assumed and the others are found by means of diagrams as we have already seen. These diagrams show only the dimensions of the inside and outside laps and travel of valve; the other dimensions of the valve and seat must be calculated.

Area of Steam Pipe. Pipes that supply the steam chest should be large enough to prevent an excessive loss of pressure due to friction. If the pipes are long they should be of such size that the mean velocity of steam in them does not exceed 100 feet per second or 6,000 feet per minute. For this calculation it is usual to assume steam admitted to the cylinder during the whole stroke.

For example. Suppose an engine is 10" $\times$ 18", and makes 180 revolutions per minute. What is the diameter of the steam pipe?

The piston displacement or volume of the cylinder is :

$$\frac{\pi d^2}{4} \times l = \frac{3.1416 \times 10^2}{4} \times 18 = 1413.72 \text{ cubic inches.}$$

$$\frac{1413.72}{1728} = .818 \text{ cubic feet.}$$

If the engine makes 180 revolutions it would use $2 \times 180 \times .818 = 294.48$ cubic feet per minute.

$$\text{The area would be} = \frac{294.48}{6000} = .04908 \text{ sq. ft.} = 7.0675 \text{ sq. in.}$$

The diameter corresponding to 7.0675 square inches is 3 inches.

A three-inch pipe would be large enough, especially if the engine cut off early in the stroke.

For a very large engine cutting off early, the allowable velocity may be taken as 8,000 feet per minute instead of 6,000 feet.

Width of Steam Port. The port opening at admission should give nearly as great an area as the steam pipe in order to prevent loss of pressure due to wire-drawing, but the actual width of the port should be great enough for the free exhaust of steam. It is well to have the steam port a little larger than the area of the steam pipe, then with a port opening of .6 to .9 of the port area for admission and full port opening at exhaust, satisfactory conditions will result.

The length of the ports is usually made about .8 the diameter of the cylinder. Then in the 10" $\times$ 18" engine the steam ports would be 8 inches long. If the area for admitting steam is 8.0675 square inches and the length of port is 8 inches, the width will be
$$= \frac{7.0675}{8} = .8834 \text{ inch, or about } \frac{7}{8} \text{ inch.}$$

The width of port opening would be about $.9 \times .8834 = .79506$ inch or about $\frac{13}{16}$ inch.

Width of Exhaust Port. When the slide valve is at its maximum displacement, the valve overlapping the exhaust port as shown in Fig. 7 reduces the area more or less. In designing the valve, the exhaust port should be of such a width that the maximum displacement of the valve does not reduce the area of the exhaust port to less than the area of the steam port. It is not advisable to make the exhaust port too large for this increases the size of the valve and thus causes excessive friction.

The height of the exhaust cavity should never be less than the width of the steam port, and may be made much higher to advantage.

Width of Bridge. The bridge must be of sufficient width so that outside edges of the valve cannot uncover the exhaust port. The width of the steam port plus the width of the outside lap plus the width of the bridge must be greater than the maximum displacement.

The width of the bridges should be not less than the thickness of the cylinder wall in order to make a good casting.

The Point of Cut-off. In the study of Indicators, it was shown that if the point of cut-off is early, the other events are not good. If a plain slide valve is used with an automatic cut-off, the cut-off

is hastened either by changing the eccentricity or by changing the angular advance. Either of these methods will accomplish the result at the expense of the compression which consequently will be earlier and excessive. Except for locomotives and high-speed engines, where compression is an advantage, the plain slide valve is not arranged to cut-off earlier than $\frac{1}{2}$ or $\frac{2}{3}$ stroke. If an earlier cut-off is desired, large outside laps are necessary. The cut-offs may be equalized by giving the head end a greater lap than the crank end, but this will cause an inequality of lead.

Lead. The lead of stationary engines varies from zero to $\frac{3}{8}$ inch according to the style of engine. An engine having high compression that compresses the steam nearly to boiler pressure, will give good results with little or no lead. If the ports are small, and the clearance large, there should be considerable lead in order to insure full initial pressure on the piston at the beginning of the stroke. Valves that open slowly require more lead than quick-acting valves.

Let us design and lay out the valve and valve seat for the following engine:

Diameter of cylinder = 10 inches.

Stroke = 18 inches.

Revolutions = 180 per minute.

Lead angle = 3° .

Cut-off to be equal at both ends and to take place at .75 stroke.

Max. port opening = .9 area of steam pipe.

Compression to be .85 of the stroke at both ends.

Length of connecting rod = 3 feet.

The piston displacement, or cylinder volume, will be $\frac{3.1416 \times 10^2}{4} \times 18 = 1413.7$ cubic inches or .818 cubic feet. If the engine makes 180 revolutions, it will use $2 \times 180 \times .818 = 294.48$ cubic feet of steam per minute. Steam pipe area = $\frac{294.48}{6000} = .0491$ square feet = 7.07 square inches.

This 7.07 square inches would also be the least possible area of the steam ports. If the length of port is made .8 the diameter of cylinder, the width will be $\frac{7.07}{8} = .88$ inches or about $\frac{7}{8}$ inch. The width of maximum port opening will be $.9 \times .88 = .792$ or nearly $\frac{13}{16}$ inch.

It will be necessary to draw a separate valve circle for each end of the cylinder. First consider the head end.

The valve travel not being known, we shall lay off XY on an assumption of 6 inches travel and draw the eccentric circle as shown in Fig. 30. Lay off the lead angle $XOA = 3^\circ$. Lay off $XC' = .75$ of the assumed valve travel = $4\frac{1}{2}$ inches. Draw the arc CC' as previously explained and draw OC which will be the crank angle at cut-off. The radius of the arc C'C will be equal to

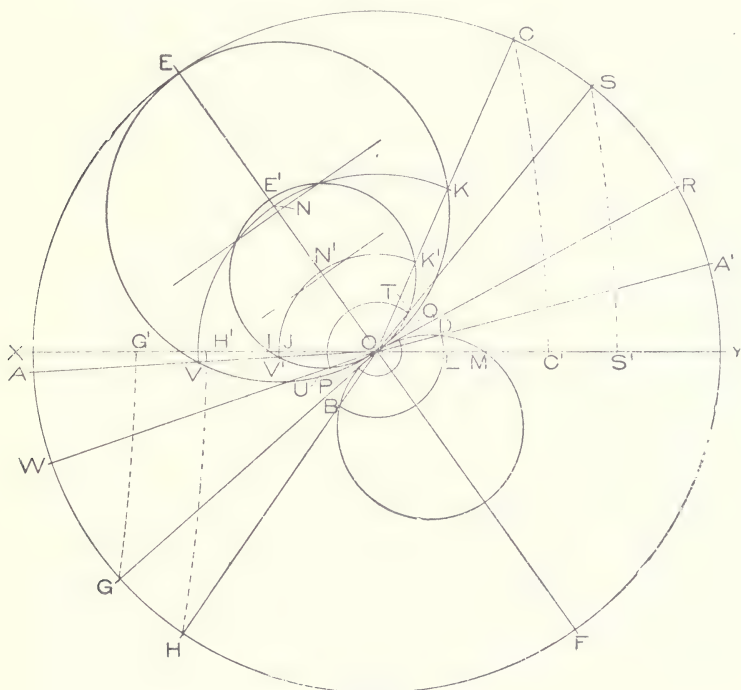


Fig. 30.

4 times the radius of the eccentric circle, or 12 inches, because the connecting rod is 4 times the length of the crank. Bisect the angle AOC by the line OE, and on OE draw the valve circle. OV = OK is then the outside lap, with these assumed condi-

tions. Draw the lap circle; then EN will be the maximum port opening. $EN = 1\frac{7}{16}$ inches, while $\frac{1}{8}$ inch is all that is necessary. The assumed eccentricity is 3 inches, therefore the probable eccentricity $= e : 3 :: \frac{1}{8} : 1\frac{7}{16}$. $e = 1\frac{1}{16}$ inches.

Now draw a new eccentric circle with a radius of $1\frac{1}{16}$ inches and a new valve circle with $OE' = 1\frac{1}{16}$ inches as a diameter. OK' is now the outside lap and the maximum port opening is equal to $E'N'$, which from actual measurement is found to be $\frac{1}{8}$ inch. The outside lap $= OK' = OV' = \frac{3}{32}$ inch and the lead is $IJ = \frac{3}{32}$ inch.

Produce EO to F and draw another valve circle. We shall use this valve circle to determine the outside laps and lead for the crank end of the cylinder. Since the cut-off is to be .75 of the stroke, we may lay off $OII' = OC'$, and with a radius of 12 inches

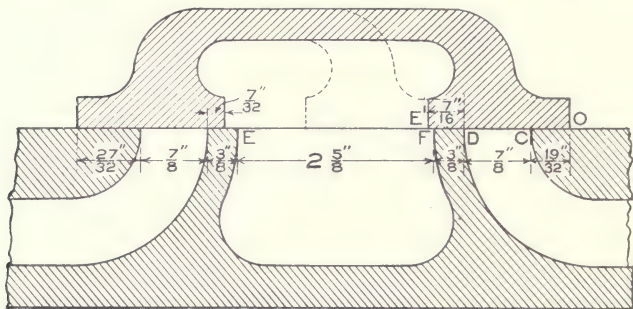


Fig. 31.

draw the arc HH' . Then, as already explained, OH will be the crank angle at cut-off on the return stroke. OB will be the outside lap $= \frac{1}{32}$ inch. Draw the lap circle intersecting the valve circle at D . Then ODA' is the crank angle at admission on the return stroke and $LM = \frac{3}{8}$ inch is the lead on the crank end of the cylinder. The maximum port opening will always be greater at the crank end than at the head end because the crank end lap is less in order to get the equal cut-off. If the laps were equal, of course the port openings would be equal.

Now lay off $YG' = .85$ of XY and find the crank position OG . This is the compression on the head end of the cylinder and gives an inside lap on this end of $\frac{7}{32}$ inch, which is equal to OP .

Draw the lap circle PQ, which allows us to draw through Q the crank line OR, which is the release on the forward stroke.

Lay off $XS' = YG' = .85$ of XY, and construct the crank line OS, which is the crank position at the crank end compression. OS intersects the valve circle at T, which gives $OT = \frac{7}{16}$ inch = inside lap on the crank end. Draw this lap circle, which will intersect the valve circle at U. This enables us to draw OUW, the crank angle at release, on the return stroke.

From the data determined by means of these diagrams the valve may now be laid out. For convenience let us tabulate the results obtained as follows:

Data.	Head End.	Crank End.
Cut-off, per cent of Stroke	75	75
Outside Lap	$\frac{27}{32}$ "	$\frac{19}{32}$ "
Inside Lap	$\frac{7}{32}$ "	$\frac{7}{16}$ "
Lead	$\frac{3}{32}$ "	$\frac{3}{8}$ "
Port Opening	$\frac{13}{16}$ "	$1\frac{1}{16}$ "
Width of Port	$\frac{7}{8}$ "	$\frac{7}{8}$ "

Fig. 31 shows this valve in section. Let us begin at the end having the largest inside lap, or in this case at the crank end. Lay out the steam port $\frac{7}{8}$ inch wide, and the crank-end outside lap = $\frac{19}{32}$ inch. The bridge will be, say, $\frac{3}{8}$ inch wide. From the inner edge of the steam port, lay off the crank-end inside lap = $\frac{7}{16}$ inch. When the valve moves to the left, the point E' will travel $1\frac{1}{16}$ inches, a distance equal to the eccentricity, and in this position of extreme displacement the exhaust port EF must be open an amount at least equal to the steam port, $\frac{7}{8}$ inch. Therefore we lay off EF equal to $1\frac{1}{16}$ " + $\frac{7}{8}$ " = $2\frac{9}{16}$ ". The inside lap overlaps the bridge nearly $\frac{1}{8}$ inch, so that we shall have to make the exhaust port opening equal to $2\frac{5}{8}$ inches. Lay off $\frac{3}{8}$ inch again for the bridge and measure back $\frac{7}{32}$ inch, equal to the head-end inside lap. The port is $\frac{7}{8}$ inch wide, and the head-end inside lap of $\frac{27}{32}$ inch completes the outline of the valve seat.

VALVE SETTING.

The principles of valve diagrams are useful in setting valves as well as in designing them. The valve is usually set as accurately as possible, and then, after indicator cards have been taken,

the final adjustment can be made to correct slight irregularities.

The slide valve is so designed that the laps cannot be altered without considerable labor, and the radius of the eccentric, which determines the travel of the valve, is usually fixed. The adjustable parts are commonly the length of the valve spindle and the angular advance of the eccentric.

By lengthening or shortening the valve spindle, the valve is made to travel an equal distance each side of the mid-position. Moving the eccentric on the shaft makes the action of the valve earlier or later as the angular advance is increased or decreased.

To Put the Engine on the Center. It is usual to put the engine on center before setting the valve. First put the engine in a position where the piston has nearly completed the outward

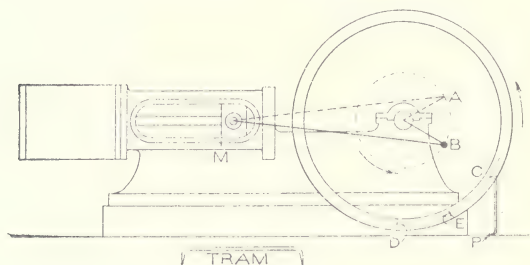


Fig. 32.

stroke, and make a mark *M* on the guide opposite the corner of the crosshead or at some convenient place. Also make a mark, with a center punch, on the frame of the engine near the crank disc or on the floor. With this punch mark *P* as a center, describe an arc *C* on the wheel rim, with a tram. A tram is a steel rod with its ends bent at right angles and sharpened.

Turn the engine past the center until the mark on the guide again corresponds with the corner of the crosshead, and make another mark *D* on the wheel with the tram, keeping the same center. With the center of the pulley or crank disc as a center, describe an arc *CD* on the rim, which intersects the two arcs drawn with the tram. Bisect the arc *CD* on the rim, included between the two short arcs, and turn the engine until the new point *E* is at

a distance from the point on the frame equal to the length of the tram, in which position the engine will be on the center.

The engine should always be moved in the direction in which it is to run so that the lost motion of the wrist pin and crank pin will be taken up the right way. In case the engine has been moved too far at any time, it should be turned back beyond the desired point and brought up to that point while the engine is moving the right way.

To Set the Valve with Equal Lead. Set the engine on the dead point and give the eccentric the proper angular advance. Adjust the length of the valve spindle to give the proper lead for that end. Now place the engine on the other dead point and measure the lead at that end. If the leads are unequal, correct half the error by changing the length of the valve spindle and the other half by altering the angular advance. In case the valve gear has a rocker, the length of the spindle should be such that the rocker will move as designed. The angular advance should not be changed, but the equal lead should be obtained by means of the valve spindle or the eccentric rod.

Second Method. In case it is difficult to turn an engine the following method may be used. First loosen the eccentric on the shaft and turn it around until it gives maximum port opening first at one end and then at the other. If the maximum port openings are not equal, make them so by changing the length of the valve spindle by half the difference. When the above adjustment has been made, set the engine on dead center and give the valve the proper lead by turning the eccentric on the shaft. The angular advance is thus adjusted.

To Set the Valve for Equal Cut-off. Place the engine on the dead point, give the eccentric the proper angular advance and the valve the proper lead. Move the engine forward until cut-off occurs, then measure the displacement of the crosshead from the beginning of the stroke. Continue moving the engine forward, until cut-off takes place on the return stroke and measure the displacement of the crosshead from the beginning of this stroke to this point.

In case the cut-off is earlier at the crank than at the head-end, the valve spindle is too short. Adjust the length of the spindle

so that the inequality will be corrected. Now set the engine on the dead point again and give the valve the proper lead by means of the eccentric. By repeating the process, making slight changes, the desired result will be obtained.

MODIFICATIONS OF THE SLIDE VALVE.

The ordinary slide valve is suitable for small engines; but for large sizes some method must be employed to balance the steam pressure on the back of the valve. With large valves, such for instance as those of locomotives or large marine engines, a great force is exerted by the steam, and the valve is forced against its seat so hard that a large amount of power is necessary to move it. This excessive pressure causes the valve to wear badly and is a dead loss to the engine. The larger the valve, the greater this loss will be.

Piston Valve. To prevent excessive pressure on the back of the valve, the piston valve is commonly used, especially in marine engines. This valve consists of two pistons, which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve. These pistons are secured to the valve stem in an approved manner and are fitted with packing rings.

The valve seat consists of two short cylinders or tubes accurately bored to fit the pistons of the valve. The port openings are not continuous as in the plain slide valve, but consist of many small openings, the bars of metal between these openings preventing the packing rings from springing out into the ports.

Steam may be admitted to the middle of the steam chest and exhausted from the ends or vice versa. With the former method, the live steam is well separated from the exhaust, and the valve-rod stuffing box is exposed to exhaust steam only. This is a good arrangement for the high-pressure cylinder; if used for a cylinder in which there is a vacuum, air may leak into the exhaust space through the valve-rod stuffing box. With this arrangement the steam laps must be inside and the exhaust laps on the outside ends.

The piston valve may be laid out and designed by means of the Zeuner diagram just as if it were a plain slide valve, and the action is the same except that it is balanced so far as the steam

pressure is concerned; the power to drive it being only that necessary to overcome the friction due to the spring rings.

Fig. 33 shows a section of the piston valve and the high-pressure cylinder for one of the engines of the U. S. S. "Massachusetts." This valve consists of two pistons connected by a sleeve through which the valve rod passes. This valve rod is prolonged to a small balancing piston, placed directly over the main

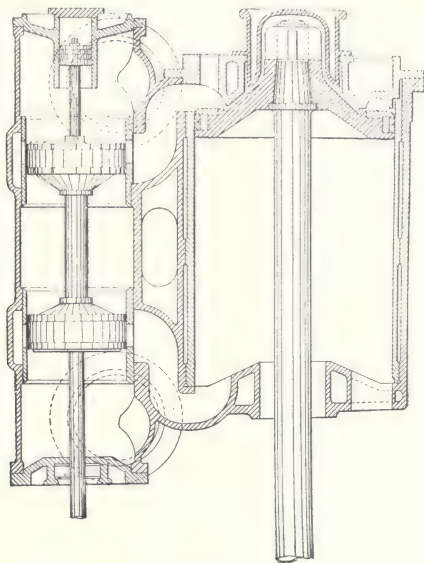


Fig. 33.

valve. The upper end of the balancing cylinder does not admit steam, so that the steam pressure below the balancing piston will practically carry the weight of the piston valve, thus relieving the valve gear and making the balance more nearly complete.

Double-Ported Valve. Sometimes it is impossible to get sufficient port opening for engines of large diameter and short stroke, especially those having a plain slide valve with short travel.

This difficulty may be overcome by means of the double-ported valve shown in Fig. 34. It is equivalent to two plain slide valves, each having its laps. The inner valve is similar to a plain slide valve except that there is communication between the exhaust space and the exhaust space of the outer valve. Each passage to the cylinder has two ports; a bridge separates the exhaust of the outer valve from the steam space of the inner valve, and the outer valve is made long enough to admit steam to the inner valve.

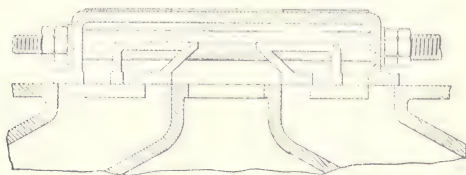


Fig. 34.

This valve may be considered as equivalent to two equal slide valves of the same travel, each having one-half the total port opening. To admit the same amount of steam as a plain slide valve, the double-ported valve requires but half the valve travel; this is advantageous in high-speed engines.

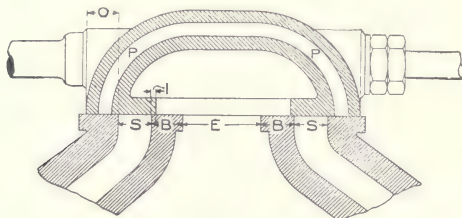


Fig. 35.

To balance the excessive steam pressure, the back of the valve is sometimes provided with a projecting ring which is fitted to a similar ring within the top of the valve chest. These rings are planed true, and fit so that steam is prevented from acting on the back of the valve. The space inside the rings is sometimes placed in communication with the condenser.

The Trick Valve. The defect of the plain slide valve, due to the slowness in opening and closing, is largely remedied in the trick valve, which is so made that a double volume of steam enters during admission. Thus a quick and full opening of the port is obtained with a small valve travel.

In Fig. 35 the valve is shown in mid-position. It is similar to a plain slide valve except that there is a passage PP through it. It has an outside lap O and an inside lap I. The seat is raised and has steam ports SS, bridges BB, and exhaust port E. If the valve moves to the right a distance equal to the outside lap plus the lead, it will be in the position shown in Fig. 36. Steam will be admitted at the extreme left edge of the valve just the same as though it were a plain slide valve; also, since steam surrounds the valve it will be admitted through the passage as shown in Fig. 36.

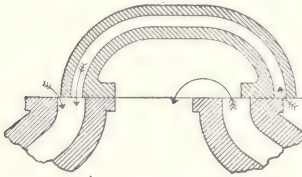


Fig. 36.

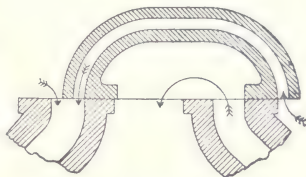


Fig. 37.

If the lead is the same as for a plain slide valve, $\frac{1}{16}$ inch for instance, this valve would give double the port opening, that is $\frac{1}{8}$ inch, when the valve was open a distance equal to the lead.

Fig. 37 shows the valve when it is in its extreme position to the right and the port is full open to steam.

Piston valves are also made with a passage similar to that of the trick valve for double admission. The valve used with the Armington and Sims engine is perhaps the best example.

Balanced Valves. Since there is a wide difference between the pressure of admission and exhaust, there must always be a great pressure acting upon the valve, causing it to run hard and wear excessively. The greater the steam pressure, the lower the pressure at exhaust and the larger the valve, the greater this pressure will be.

Piston valves are commonly used on the high and intermediate cylinders of triple-expansion engines, and if well made and fitted with spring rings, should not leak. Small piston valves are often made without packing rings; but even if they fit accurately when new, they soon become worn and cause trouble.

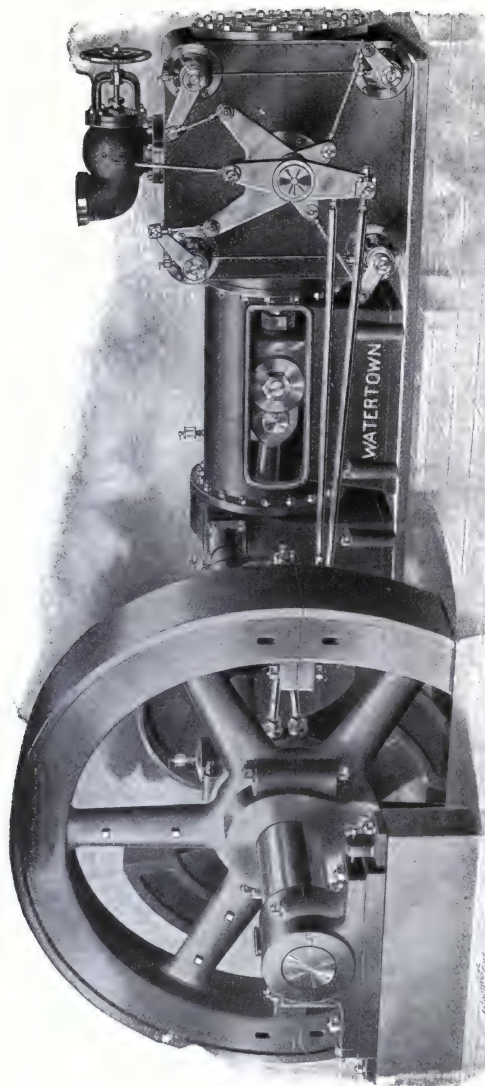
The double-ported valve, the trick valve, and others often have some device for relieving the pressure, such as a bronze ring or cylinder, fastened to the back of the valve. This ring is pressed by springs against a finished surface of the valve chest cover, and the space thus enclosed by the ring may be connected to the exhaust. There are numerous devices for balancing valves, but they are usually more or less expensive and are liable to cause trouble from leakage.

STEPHENSON LINK MOTION.

One of the earliest, and at present one of the most common mechanisms for reversing engines, or changing the ratio of expansion, is the Stephenson link motion, shown in Fig. 38. This illustration is taken from the drawings of a recent battleship engine, and may be considered the typical arrangement of the Stephenson gear as applied to marine practice.

The two eccentrics E and E' , whose centers are at C and C' , respectively, are shown in their relative positions when the crank OA is at dead center. The eccentric rods R and R' are connected by forked ends to the link pins H and G . The link consists of two curved bars bolted together in such a manner that they may slide by the link block N . On the link are three sets of trunnions; the two outer ones, or link pins, are fitted into the forked end of the eccentric rods, and the middle one, known as the saddle pin, is fitted into the end of the drag links FM .

The valve stem has, at its lower end, a pivoted block N , called the link block, provided with slotted sides through which the links can slide from right to left. The reverse shaft, or rock shaft, K , here shown in full gear "forward," may be turned until F moves over to B ; in this position the link will be pushed across the link block, and the valve will get its motion from the rod R' instead of from R as before. The link in this position would be full gear "astern."



HIGH SPEED CORLISS ENGINE.
Watertown Engine Co.

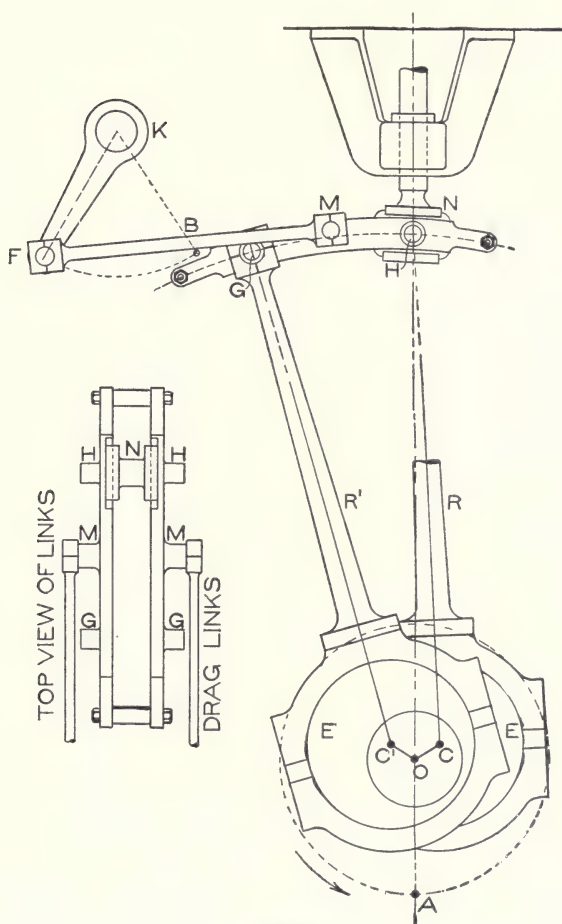


Fig. 38.

In all large engines, such as marine, the reverse shaft is turned by power, but in smaller engines, such as locomotives, the engineer can turn the shaft by means of a lever.

When set full gear forward, as in Fig. 38, the valve admits steam to the crank end of the cylinder, and the crank revolves as

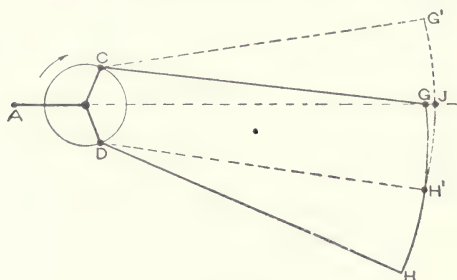


Fig. 39.

shown by the arrow. As the crank turns, both eccentrics impart motion to the link, but the "go ahead" link pin II approximately coincides with the link block, so that nearly all its up-and-down motion is transmitted to the valve stem, while the "go astern"

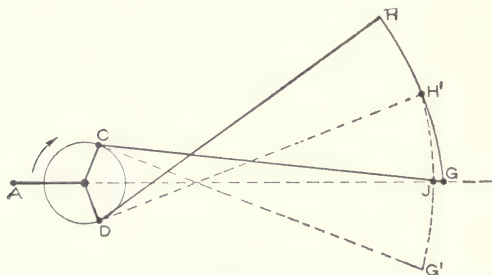


Fig. 40.

eccentric exerts but little effect upon the link block. Moving the drag links over to the extreme right reverses all these conditions by bringing the other link pin under the link block. In this position, steam will be admitted to the other end of the cylinder, and the engine will run in the opposite direction. This will be clearly seen by referring to Fig. 38.

When at full gear, either forward or backing, the valve moves as if there were really but one eccentric, while at intermediate points its motion is the result of the combined influence of both eccentrics, one tending in a measure to counteract the other. The effect of this is to shorten the valve travel the same as if the valve were driven by a new eccentric having less throw than either of the other two.

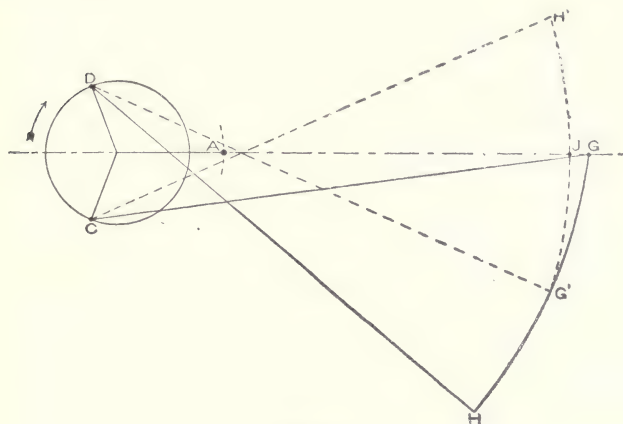


Fig. 41.

Decreasing the valve travel causes cut-off to occur earlier, compression is earlier, release later, and the lead is reduced somewhat. If every point of the link moved in the arc of a circle when the drag link is shifted, the lead would not alter; but, since the eccentric rods about which each end swings are centered at different points, C and C', this is impossible.

Figs. 39 and 40 show the two principal ways of arranging the eccentric rods of a Stephenson gear. The first is said to have "open rods", the second "crossed rods"; referring to whether the rods are crossed or open when both the eccentrics face the link. It can easily be seen that when the eccentrics shown in Fig. 39 have turned through 180° they will be in the position shown in Fig. 41, but this is the same arrangement as before and is "open" rods. The full lines show the positions in full gear forward, while

the dotted lines indicate the positions in mid gear. With open rods it will be seen that when at full gear the link block is at G, and that if, without turning the crank, the link is shifted to mid gear, then the link block moves to J, Fig. 39, and the valve must consequently be moved toward the right an amount equal to GJ, thereby increasing the lead on the crank end of the cylinder. With crossed rods, moving the link from full to mid gear moves the link block from G to J, Fig. 40, thus reducing the lead. It follows then that open rods give increasing lead from full toward mid gear, and that crossed rods give decreasing lead. With crossed rods there will be no lead when in mid gear. It will be apparent that the shorter the rods the greater this increase or decrease will be.

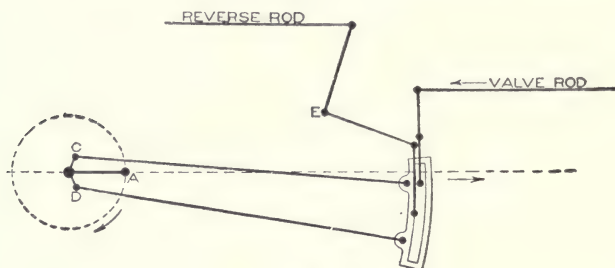


Fig. 42.

Nearly all marine engines, and some English locomotives, have their link blocks carried directly on the valve rod. American locomotives commonly use a rocker, one end of which carries the link block while the other moves the valve rod. This arrangement indicated in Fig. 42 makes it possible to place the valve and steam chest above the cylinder. The position of the crank for the same valve position is just opposite that shown in Fig. 39 because the rocker reverses the valve motion; this gives an arrangement of crank and eccentrics that is identical with that indicated in Fig. 41 and the rods, although apparently crossed, are in reality of the open rod arrangement, giving increasing lead toward mid gear. A rod from the bell-crank lever on the reverse shaft E, leads back to the engineer's cab and connects with the reverse lever. This lever moves over a notched arc, and may be held by a latch in any

one of the notches, thus setting the link in any position from mid gear to full gear, either forward or back.

The Stephenson link is designed to give equal lead at both ends of the cylinder; but to accomplish this, the radius of the link arc (that is an imaginary line in the center of the slot) must be equal to the distance from the center of this slot to the center of the eccentric. In Fig. 38 the radius of the link arc is equal to CH and C'G.

Exact quality of lead is not essential, and the radius of the link arc is sometimes made greater or less than stated above in order to aid in equalizing the cut-off; but the change should never be great enough to affect the leads.

Stephenson originally intended to use the link simply as a reversing gear, but soon found, however, that at intermediate points between the two positions of full gear, it would serve very well as a means of varying the expansion and cut-off. Very soon the link came to be used not only on locomotives and marine engines, but on stationary engines as well, in connection with the reverse shaft which was under the control of the governor. The mechanism proved to be too heavy to be easily moved by a governor and it has gradually fallen into disuse on stationary engines excepting as a means of reversing.

In marine practice, the variable expansion feature is of little value, for marine engines run under a steady load and the link is set either at full gear or at some fixed cut-off. For locomotives, however, the variable expansion is nearly as important as reversing. Locomotives are generally started at full gear, admitting steam for nearly the entire stroke, and then exhausting it at relatively high pressure. This wasteful use of steam is necessary to furnish the power needed in starting a train. After the train is under way, less power is required per stroke, and the link is gradually moved toward mid gear, or "notched up" by the engineer, thus hastening the cut-off; the expansion is increased and the power is reduced in proportion to the load.

As the cut-off is changed, it is desirable to maintain an approximately equal cut-off at each end of the cylinder; this can be secured in the Stephenson gear by properly locating the saddle pin and the reverse shaft. When used without a rocker, as in

Fig. 38, the saddle pin should be on the arc of the link or slightly ahead of it. When used with a rocker, the saddle pin should be behind the link arcs, and to give symmetrical action for forward and backward running, it should be opposite the middle of the arc, that is, equally distant from each link pin.

The Stephenson link cannot be designed directly from the Zeuner diagram, but a systematic investigation can be made by using a wooden model of the proposed link. This can be mounted on a drawing board, and the effect of changing the position of pins and the proportions of rods and levers can be determined without difficulty. By a system of trials a combination can be found best suited to obtain the desired results. Moreover, a model makes it possible to measure directly the slip of the link block along the link. This slip should be kept as small as possible to prevent rapid wear. It can be controlled to some extent by properly locating the link pins, by avoiding too short a link, and by choosing a favorable position for the reverse shaft.

The Gooch Link. Another form of link motion, known as the Gooch Link, is illustrated in Fig. 43. It has been extensively used on European locomotives, although it is gradually being replaced by a type of valve gear known as the Walschaert, which will be described later.

The Gooch link has its concave side turned toward the valve instead of toward the eccentric. The radius of curvature of the link is equal to AB , the length of the radius rod. The link is stationary and the link block slides in the link. The engine is reversed by means of the bell-crank lever on the reverse shaft E which shifts the link block instead of the link, as is the case with the Stephenson. The link is suspended from its saddle pin M , which is connected by a rod to the fixed center F , so that the link can move forward and back as the eccentricity is changed, or it can pivot about its saddle pin as the eccentrics revolve.

Since the radius of the link arc is equal to AB , it is apparent that the block can be moved from one end of the link to the other, that is, from full gear "forward" to full gear "back" without moving the point A , which is on the end of the valve rod. The lead then is constant for all positions of the block, and the distribution of steam for locomotives is slightly preferable to that

obtained by the Stephenson; but the gear is more complicated and requires nearly double the distance between shaft and valve stem.

The variable lead is perhaps a slight advantage to the locomotive, which is a slow-speed engine in starting, thus requiring but little lead. As the speed increases, and the link is "notched up", the lead is increased as the cut-off is shortened, and at high speed we have a large lead. With the Gooch link, the lead can be set for the average running speed, and although a little too great for good work at slow speed, it is a matter of small consequence, because the engine runs at slow speed but a very small fraction of the time it is in service, and the loss due to large lead at slow speed is of no consequence whatever in a day's run.

Several other link motions have been used; but at the present time probably more Stephenson link motions are used than all

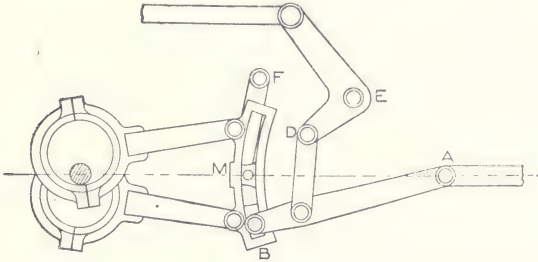


Fig. 43.

the other forms of reversing gear combined, and when a "link motion" is mentioned, the Stephenson is usually meant unless otherwise specified.

RADIAL VALVE GEARS.

In general, it would be desirable to have precisely similar steam distribution at each end of the cylinder, and it would often be of great advantage with an expansion gear like the Stephenson, if the cut-off could be shortened without changing any other event of the stroke. A Stephenson gear can be made to maintain equality of lead for both ends of the cylinder as the cut-off is shortened, but we have seen that in so doing, the lead of both ends is either

increased or diminished according as the link is arranged with "open rods" or "crossed rods". Moreover, the compression is hastened by bringing the link to mid-gear, all of which in many instances is undesirable.

This disadvantage of the Stephenson link motion lead to the design of the so-called "Radial Valve Gears", many of which are so complicated as to be impracticable, but all of which obtain a fairly uniform distribution of steam.

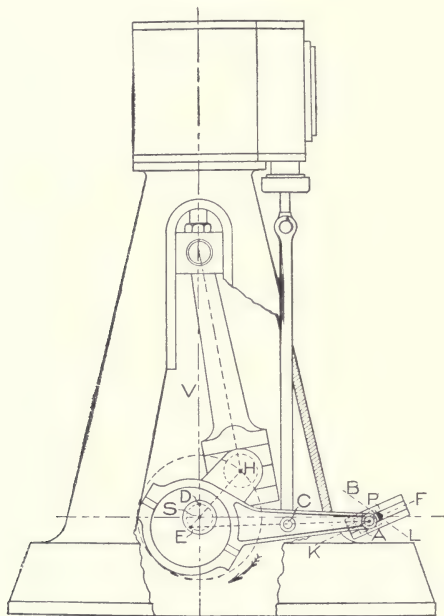


Fig. 44.

Hackworth Gear. The essential features of the Hackworth Gear are indicated in outline in Fig. 44. In this figure, S is the center of the shaft, and the eccentric E is set 180° from the crank SH. At the right-hand end of the eccentric rod EA, is pivoted a block which slides in a straight, slotted guide. The guide remains stationary while the engine is running, but can be turned on its

axis P, to reverse the engine or change the cut-off. P is a pivot, located on the horizontal through S in such a position that $DP = EA$. If these two distances are equal, A will coincide with P when the crank is at either dead point and the slotted guide may be turned from "full gear forward", as shown in the figure, through the horizontal position to "full gear backing", as

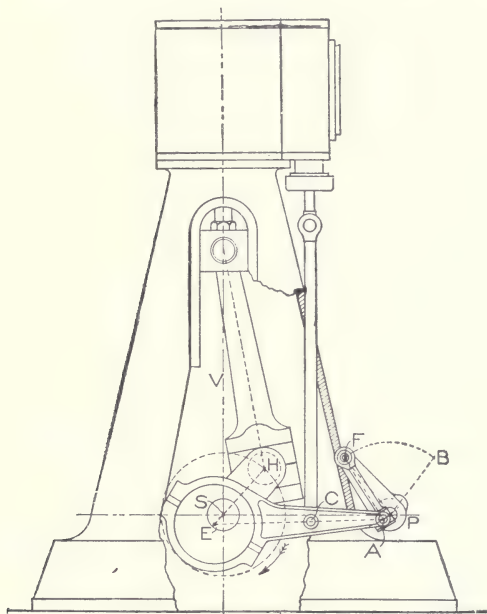


Fig. 45.

shown by the line BL, without moving the valve. Therefore the leads are constant for all positions of the guide. The valve rod running upward from C, connects with the valve stem which it moves in a straight line. The valve stem is made just long enough to equalize both leads, and if the point C has been properly chosen, the two cut-offs will be very nearly equal for all grades of the gear.

A somewhat better valve action is obtained by slightly curving the slotted guide, with its convex side downward. This gear is sometimes used on marine engines and on small stationary engines.

Marshall Gear. The most objectionable feature of the Hackworth gear, is the slotted guide, for the sliding of the block causes considerable friction and wear. The Marshall gear, shown in outline in Fig. 45, is designed to obviate this feature. The point A moves in the desired path by swinging on the rod FA about F as a center. While the engine is running, the lever FP remains stationary, but can be turned on its axis P to reverse the engine, or change the cut-off. The pivot P, is located precisely as in the Hackworth gear, and the lever FP can be turned from "full gear forward", as shown in the figure, to "full gear backing", as shown by the line BP, intermediate positions give different cut-offs as with the Hackworth gear. Since FA is made equal to FP, the point A will always swing through P, no matter where F may be, and will coincide with P, when the engine is on dead center. The leads therefore will remain constant, as in the preceding case.

The Marshall gear is sometimes made with C at the right of A on a prolongation of the line EA. In this case if the same kind of valve is to be used, the eccentric E must move with the crank instead of 180° from it. The Marshall gear is frequently used on marine engines, the one eccentric being simpler than the two required by the Stephenson.

Joy Gear. Perhaps the most widely known, and certainly one of the best radial gears is the Joy, outlined in Fig. 46. It is frequently used on marine engines and on some English locomotives. No eccentrics are used, the valve motion being taken from C, a point on the connecting rod. H is a fixed pivot supported on the cylinder casting. The lever ED has a block pivoted at A, which slides back and forth in a curved slotted guide. The guide and the lever PF are fastened to the reverse shaft P, and by means of a reverse rod leading off from F, can be turned from full gear forward, as shown, to full gear backing when the pin F moves over to B. Motion is transmitted to the valve stem by means of the radius rod EG. The proportions are such that when the crank is on either dead point, the pivot of block A coincides

with P, so that the curved guide may then be set in any position without moving the valve; therefore the leads are constant. This gear gives a rapid motion to the valve when opening and closing and a more nearly constant compression than the Stephenson gear, and the cut-off can be made very nearly equal for all grades of the gear. Its many joints cause wear and its position near the crosshead, makes a careful inspection of the crosshead and piston exceedingly difficult while the engine is running.

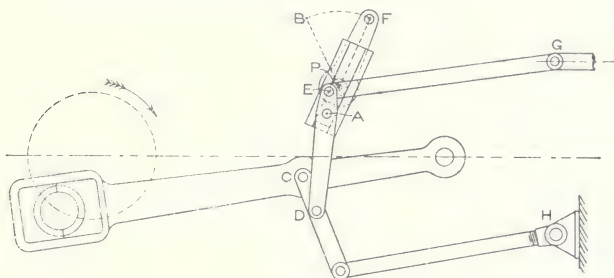


Fig. 46.

Walschaert Gear. This radial valve gear, although seldom seen in the United States, is the valve mechanism most commonly used on locomotives built on the continent of Europe. Like all other radial gears, it gives constant lead, and a distribution of steam very nearly alike for each end of the cylinder. In this respect it is superior to the Stephenson link, and gives without doubt better economy, but its mechanical construction is complicated, and not well adapted to the American type of locomotive. Fig. 47 illustrates this type of gear. S is the center of the driver axle. The crank pin K has forged on its center end an arm KE, on which the pin E is fixed. This arm lies parallel to the plane of the driving wheels, and being fixed to the crank pin, turns with the wheel, allowing the connecting rod to pass between it and the driving wheels. In this manner the point E moves around S in a circle, and moves the rod EH back and forth just as if it were an eccentric. It is so made that ES is perpendicular to the crank KS, and therefore the action of the pin E is equivalent to an eccentric with no angular advance.

This arm reaching back from the outer end of the crank pin is one of the most objectionable features on the construction, and is sometimes replaced by the regular type of eccentric put on the shaft between the driving wheels.

The eccentric rod EH causes the box link HP to oscillate on fixed trunions P. This link has a groove curved to a radius equal to GD, the length of the radius rod. A block pivoted at G, on one end of the radius rod, is free to move up or down in this

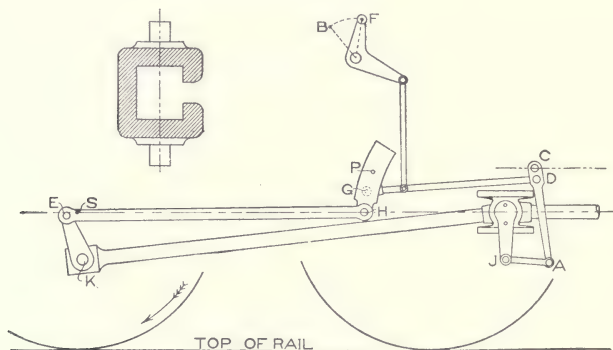


Fig. 47.

groove. The valve derives its motion from C, a pivot on the floating lever CA. Point A receives motion from the crosshead; point D from the eccentric and the curved link; and a combination of these two imparts motion to C (which can slide only along the dotted line). A bell-crank lever pivoted above the link shifts the mechanism from "full gear forward" when F is moved to B, thus raising G above the link pivot or saddle pin.

ADJUSTABLE ECCENTRICS.

The position of an eccentric for a plain slide valve is 90° plus the angular advance ahead of the crank, in the direction in which the engine is to turn. Thus A, Fig. 48, is correctly placed, relative to the crank C, if the engine turns right-handed. For running in the opposite direction, the position of the eccentric is at D. Some engines are provided with a reversing mechanism which causes the eccentric to shift from A to D, either along the arc

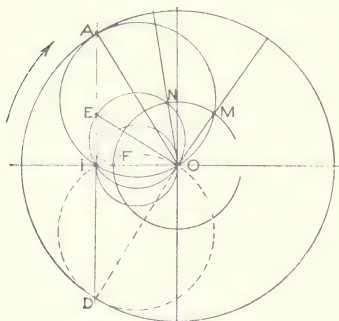


Fig. 50.

change in the lead IF . Let us consider the eccentric position OA , Fig. 48. In this position OI represents the displacement of the valve from mid-position when the engine is on center. If the eccentric moves to OB , the displacement will be OM , which is greater, showing an increase in lead equal to IM , but if the position is OE instead of OB , the displacement from mid-position will

be OI as before. It is evident that the eccentric can move on the straight line from A to D , without changing the lead, while the decreased valve travel will result in an earlier cut-off. If the

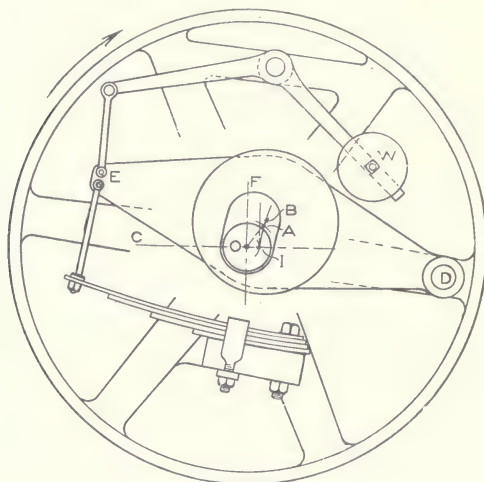


Fig. 51.

shifting eccentric is to be used for an automatic cut-off, as in the various types of fly-wheel governor, the curved path is not

desirable on account of excessive lead at short cut-off, but if it is to be used only as a means of reversing, it is preferable to the straight line.

All fly-wheel governors operate by shifting the eccentric, either to change the angular advance, the travel of the valve, or both. Fig. 51 illustrates the principle of a governor arranged to give decreasing lead, but as these mechanisms are described in the *Steam Engine—Part I*, under the head of governors, a further discussion will not be given here.

The device shown in Fig. 52 is often used for reversing engines of small launches. The eccentric *E* is loose on the shaft between a fixed collar *G*, and a hand wheel *H*. A stud projecting from the eccentric passes through a curved slot in the disc of the wheel, and can be clamped by a hand nut *F*. When running forward with the crank at *C*, the center of the eccentric is at *A*, and the nut clamped at *F*. To reverse, steam is shut off, and when the engine stops, the nut *F* is loosened, and then moved to *B* and clamped; or after *F* is loosened, the wheel, shaft, crank and propeller are turned over by hand until *B* strikes the stud at *F*, where it is clamped. The engine will then run astern.

To study the application of the Zeuner diagram to this form of mechanism turn again to Fig. 51. If *OA* is the desired eccentricity for a normal position of the governor, the perpendicular distance of *A*

from *OF* is made equal to the lap *OI*, plus the desired lead. Pivot *D* is then located equally distant from *A* and *I*. Zeuner's diagram for this gear, drawn to an enlarged scale, is shown in Fig. 53. The angular advance *F'OB* is laid back toward *OC*. *OB* is the

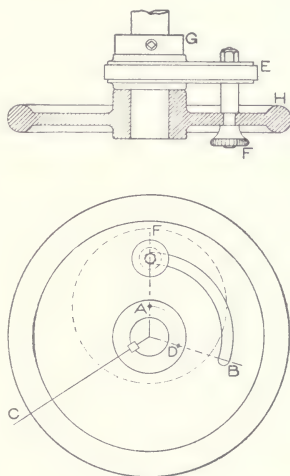


Fig. 52.

maximum eccentricity; OI , the lap or the desired least eccentricity. An arc, with proper radius, described through B and I shows the path of the eccentric. If the eccentric moves in to A , the crank angle at cut-off is decreased from COD to COE , and the lead decreased from FI to GI . A slight decrease in lead is

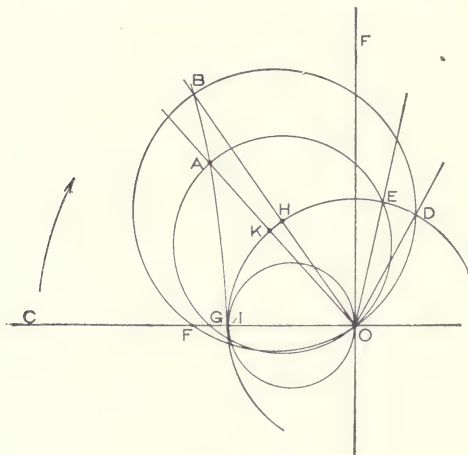


Fig. 53.

not objectionable, since the speed is not allowed to increase more than two or three per cent; and further, as the lead increases, compression decreases, so that one influence helps to counteract the other. The decrease in maximum port opening from BH to AK is unavoidable, but it is permissible, since it occurs only when the load decreases, and when less steam should be admitted to the cylinder.

DOUBLE VALVE GEARS.

It has been shown in the preceding discussion, that a plain slide valve under the control of a gear that gives a variable cut-off, such as a shifting eccentric or a link motion, will not give a satisfactory distribution of steam at short cut-off owing to excessive compression, variable lead, or early release. These difficulties are overcome in a measure by the use of the radial gear; and also by the use of two valves instead of one. The main

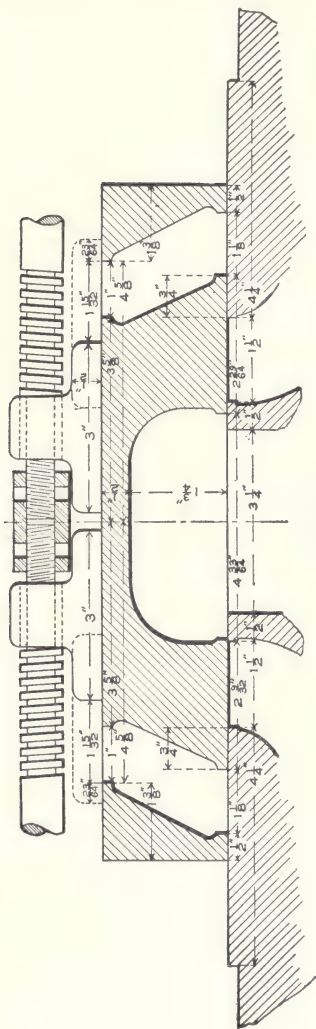


Fig. 54.

valve controls admission, release, and compression; the other, called the cut-off valve, regulates the cut-off only, which may be changed without in any way affecting the other events of the stroke. This cut-off valve may be placed in a separate valve chest, or it may be placed on the back of the main valve.

Meyer Valve. The most common form of double valve gear is the Meyer Valve, Fig. 54. The cut-off valve is made in two parts and works on the back of the main valve. The two parts are connected to a valve spindle with a right- and left-hand thread, so that their positions may be altered by rotating the valve spindle.

A swivel joint is usually fitted in the valve-spindle between the steam chest and the head of the valve rod, and the valve spindle prolonged into a tail rod which projects through a stuffing box on the head of the steam chest. See Fig. 55. The end of this tail rod is square in section and reciprocates through a small hand wheel by means of which it can be rotated while the engine is running, whatever the position of the valve may be.

Each valve is under the control of a separate eccentric. The eccentric moving the main valve is usually fixed, while the cut-off valve eccentric may be under the control of a governor. Since a slight compression is desired, the main valve is set to give late cut-off. This will give late release and late compression, and allow a wide range of cut-off for the cut-off valve. With this gear, lead, release, and compression are entirely independent of the ratio of expansion, and the cut-off is much sharper, because the cut-off valve, when closing the ports, is always moving in a direction

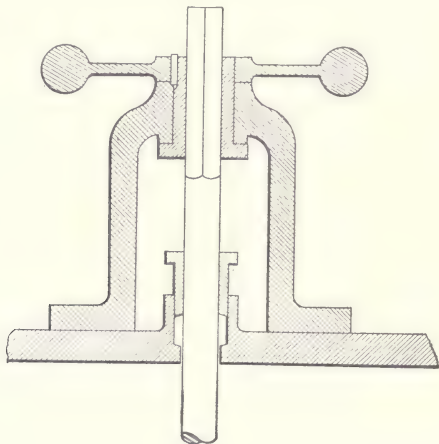


Fig. 55.

opposite to that of the main valve. The valve may be designed by means of Zeuner's diagrams.

Design of a Meyer Valve. Let us design a Meyer Valve having an eccentricity of 2 inches. Let the eccentricity of the cut-off valve be $2\frac{1}{4}$ inches and the relative travel of the cut-off valve in relation to the main valve be 3 inches. This will make the relative motion of the cut-off valve equivalent to the travel of a plain slide valve with an eccentricity of $1\frac{1}{2}$ inches. Let the outside lap on the main valve be $\frac{3}{4}$ inch, the lead $\frac{1}{32}$ inch, the compression 95 per cent of the stroke, and let the ratio of the length of the crank to connecting rod be six.

In Fig. 56 draw XOY equal to 4 inches, the main valve travel. Lay off $YD = 95$ per cent of $4 = 3.8$ inches, and with a radius of 12 inches, and the center on YX produced draw the arc $DILK$. $H.K.O$ is the crank position at compression. $C.K.O$, the crank position at cut-off, is found in a similar manner. Lay off OI equal to the lap plus the lead, and draw the valve circle for the main valve through I and O with a diameter equal to its eccentricity of 2 inches. To do this take a radius equal to 1 inch, and draw arcs from I and O as centers that shall intersect at B . B is

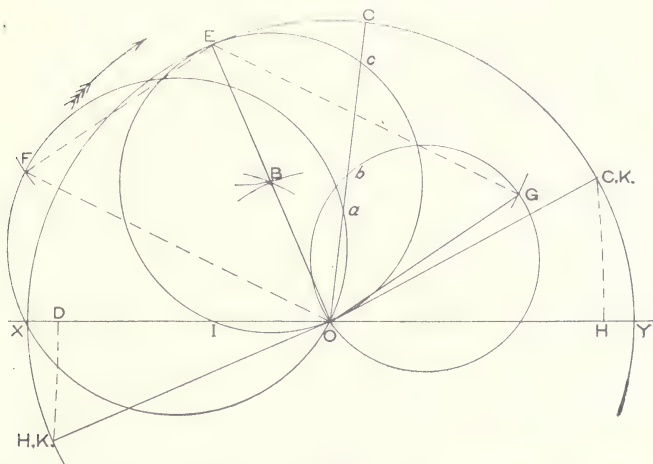


Fig. 56.

the center of the valve circle and OBE is the eccentricity, 2 inches. With E as a center, and with a radius equal to half the relative travel of the cut-off valve (in this case $1\frac{1}{2}$ inches), draw an arc. With O as a center and with a radius equal to $2\frac{1}{4}$ inches, the eccentricity of the cut-off valve, draw another arc intersecting the first one at F . On OF as a diameter construct a valve circle. This valve will represent the absolute motion of the cut-off valve, independent of the motion of the main valve. This circle then will show the displacements of the cut-off valve from the center of the steam chest. With E as a center and with a radius equal to FO draw an arc, and with O as a center and with a radius equal

to EF draw another arc intersecting the first at G. On OG as a diameter construct a valve circle. This circle will then represent a travel of the cut-off valve moving on the main valve. That is, it will represent the displacements of the cut-off valve from the center of the main valve. This circle is not, properly speaking,

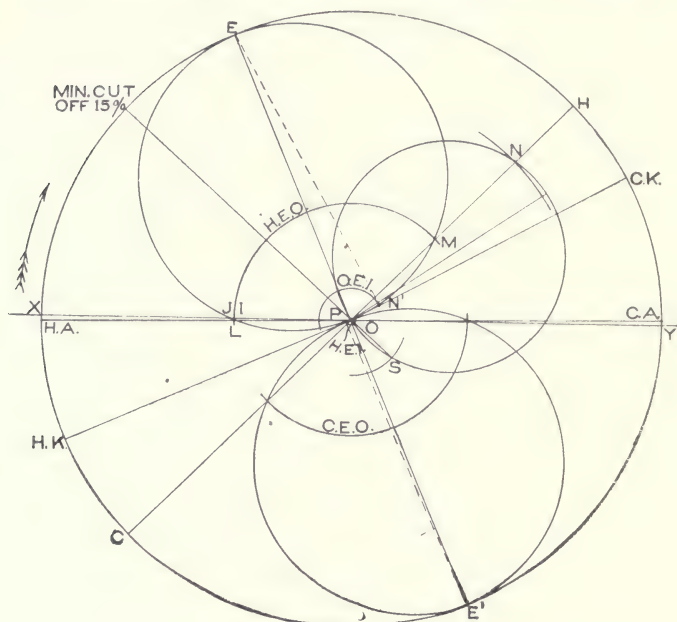


Fig. 57.

a valve circle and OG is not an eccentricity, but simply represents the relative motion of the two valves. This can be proved by analytical geometry, but an inspection of the figure shows that this must be true.

Draw the crank line OC at any position, cutting the valve circles at *a* and *b* and *c*. O*a* represents the absolute displacement of the cut-off valve, that is, from the center of the steam chest and O*c* represents the displacement of the main valve. The

relative displacement of the cut-off valve, that is, from the center of the main valve, will be the difference between Oc and Oa , since both valves are moving in the same direction. By careful measurement it will be found that $Ob = Oc - Oa$, and any arc as Ob on the auxiliary circle ObG will correctly represent the displacement of the cut-off valve from the center of the main valve at the corresponding crank angle.

Fig. 57 shows the crank angle at head-end compression H.K., and at crank-end compression C.K., the main valve circle, and the auxiliary circle which are transferred from Fig. 56. The construction lines and all lines not essential to the figure are omitted to avoid confusion.

Lay off on Fig. 57, OI equal to the outside lap $\frac{3}{4}$ inch and draw the head-end lap circle H.E.O. It will intersect the valve circle for the main valve at L and M. Through L draw the crank position at admission (head-end) H.A. and the crank position at cut-off through M. This gives the greatest possible cut-off. The cut-off valve may be set to give a much earlier cut-off than this, but of course a later setting would be of no avail for the port would be closed by the main valve at this angle. The crank line OMH cuts the auxiliary circle at N, so that ON ($1\frac{1}{3}\frac{5}{8}$ inches) is the clearance of the cut-off valve. That is, the edge of the cut-off valve must be set $1\frac{1}{3}\frac{5}{8}$ inches from the edge of the main valve port in order to cut-off at this crank angle. The full lines of Fig. 54 show the cut-off valve placed in this position.

The intersection of H.K.O with the lower valve circle, gives the inside lap at the head end of the cylinder. This line comes so nearly tangent to the valve circle that the intersection can be determined only by dropping a perpendicular to H.K.O. from E'. This cuts the circle at P and $OP = \frac{1}{3}\frac{1}{2}$ inch equals the head-end inside lap, and H.E.I. represents the corresponding lap circle.

The crank-end angle at compression is C.K. which cuts the upper valve circle at N', giving an inside lap for the crank end of $ON' = \frac{1}{6}\frac{3}{4}$ inch. To make this intersection more apparent the perpendicular can be drawn from E as previously explained.

Suppose that it was required that the minimum cut-off should be 15 per cent. Find the crank position at 15 per cent of the stroke in the same manner as the crank position was found at compression.

Produce this line through O until it cuts the auxiliary circle at S. Then $OS = \frac{23}{64}$ inch = the required lap for the cut-off valve in order to cut-off at 15 per cent of the stroke. The dotted lines in Fig. 54 show the cut-off valve drawn in this position.

For a valve of this sort, the cylinder port should be $1\frac{1}{2}$ inches wide and the valve port 1 inch wide. Fig. 54 shows this valve laid out to scale, but as this process is in all respects similar to that described for laying out a plain slide valve, it will not be described in detail.

DROP CUT-OFF GEARS.

The ordinary slide valve controls eight different events of the stroke, that is, admission, cut-off, release, and compression for both ends of the cylinder. A change in the setting of a plain slide valve that affects any one event on the crank end, let us say, will also change to a greater or less degree every other event of the stroke, on the head end as well as on the crank end; so that in setting a slide valve the desired position for one event must usually be sacrificed in order to make the others less objectionable.

In order to provide a better distribution of steam than is possible with a single valve, some engines have four valves, two at each end of the cylinder. In horizontal engines, two are placed above the center line of cylinder and two below. The upper are for admission and cut-off, the lower for release and compression. Since each valve controls but two events, a very satisfactory adjustment can be made and the extra complication and cost for large engines are more than overbalanced by the advantages gained, viz.: A very much better distribution of steam, short steam passages and small clearances, separate ports for the admission of hot steam and the exhaust of the same steam after expansion when its temperature has fallen, and finally the possibility of opening and closing the ports very rapidly, thus preventing wire-drawing. The small clearances, short ports and separate admission and exhaust materially reduce the cylinder condensation, and thus effect a large saving in the steam consumption.

When four valves are used, for high speeds, the motions of all must be positive, that is, they must be connected directly to some mechanism that either pushes or pulls them through their entire motion, but for speeds up to 100 revolutions or so a disen-

gaging mechanism may be used, and the valves may shut of themselves, either by virtue of their weight or by means of springs or dashpots. The valve is usually opened by means of links or rods, moved by an eccentric, and at the proper point of cut-off the rod is disengaged from the valve which drops shut, hence the term "drop cut-off" gears.

Reynolds-Corliss Gear. The most widely known drop cut-off gear is the Reynolds-Corliss, shown in Figs. 58 and 59; it is often referred to as the *Reynolds hook-releasing gear*. An eccentric on the main shaft gives an oscillating motion to a circular disc

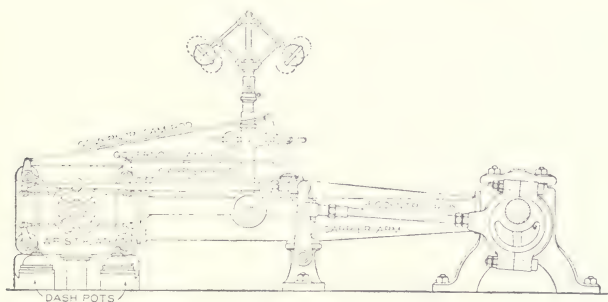


Fig. 58.

called the wrist plate, pivoted at the center of the cylinder. It transmits motion to each of the four valves through adjustable links known as *steam rods* or *exhaust rods*, according to whether they move the admission or exhaust valves.

The valves which are shown in section in Fig. 60 oscillate on cylindrical seats, and the position of the *rods* is so determined that they give a rapid motion to the valve when opening or closing, and hold it nearly stationary when either opened or closed.

The Reynolds hook is shown in detail in Fig. 59. The *steam arm* is keyed to the valve spindle which passes loosely through a bracket on which the *bell-crank* lever turns, and the spindle is packed to make a steam-tight joint where it enters the cylinder. Motion of the *steam rod* toward the right will turn the *bell-crank* lever and raise the *hook stud*. The *hook* (from which the gear derives its name) pivoted on this stud, has at one end a hard-

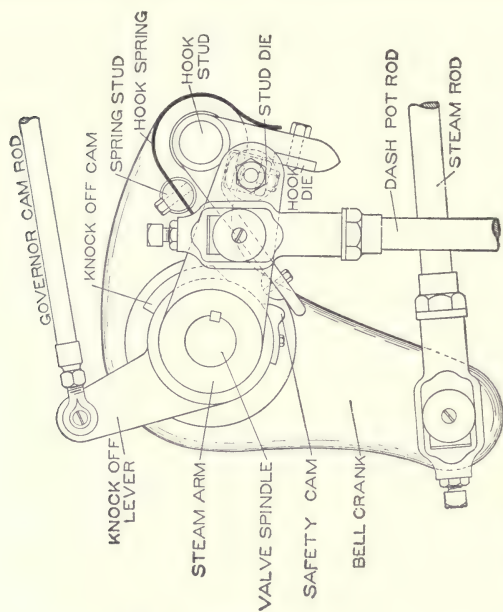
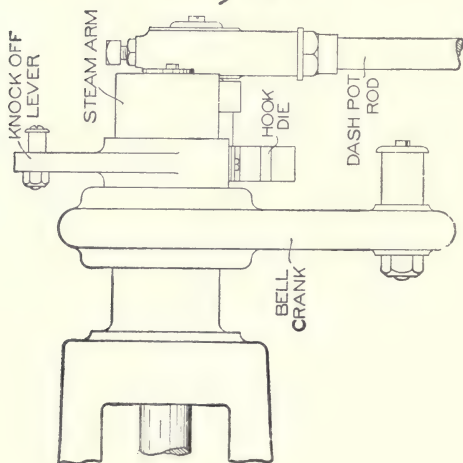


Fig. 59.



ened steel die with sharp, square edges, and at the other end, a small steel block with a rounded face. As the hook rises, the *hook die* engages the *stud die* which is fastened to the *steam arm*, and one end of the steam arm is thus raised. This turns the valve in its seat and admits steam. As the *hook* continues to rise, its stud moves in an arc above the valve spindle, and the round-faced block at its left-hand end strikes the *knock-off cam* which causes the *hook* to turn about its stud and disengage the *hook die* from the *stud die*. In raising the *steam arm*, the *dash-pot rod* also is raised and a partial vacuum is created in the dash-pot. As soon, therefore, as the dies become disengaged, the *dashpot rod* quickly drops under the force of this vacuum, thus turning the steam arm and closing the valve. The striking of the left-hand end of the hook against the knock-off cam determines the point of cut-off, by releasing the valve at that instant.

This cam is a part of the *knock-off lever* to which the *governor cam rod* is fastened. Any action of the governor which would cause the *cam rod* to move toward the right would cause this *knock-off lever* to turn on its axis, the steam arm, and consequently lower the position of the *knock-off cam*. This would cause an earlier contact between the cam and end of hook, and consequently an earlier cut-off. By lengthening or shortening the *governor cam rod*, the point of cut-off can be adjusted to suit the engine load without changing the speed.

There is a limit to this adjustment, for it can be shown that a Corliss gear operated by a single eccentric cannot be arranged to cut-off later than half stroke. Suppose the eccentric is set just 90° ahead of the crank. Then it will reach its extreme position just as the piston gets to half stroke. If by that time the *hook* which was rising and opening the admission valve, has not yet struck the *knock-off cam*, it cannot strike it at all, for any further motion will cause the hook to descend to its original position, that is its position at the beginning of the stroke; the hook will not disengage from the steam arm *stud* at all and the bell crank will return, closing the valve in the same manner in which it opened it. Cut-off will then take place near the end of the stroke, but it will not be the sharp cut-off produced by the sudden *drop* when the dies are disengaged.

If the eccentric were set less than 90° ahead of the crank, the cut-off could be arranged to occur later than half stroke, but this is decidedly impracticable, for with such a position of the eccentric the action of the valves at release and compression is spoiled. When it is necessary to cut-off later than half-stroke, as sometimes happens on low-pressure cylinders of compound engines, it may be arranged for by means of two eccentrics, one set more than 90° ahead of the crank to operate the exhaust valves, and one less than 90° ahead to operate the admission valves.

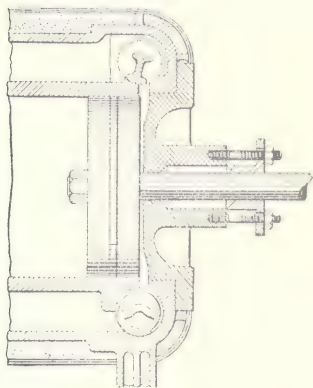


Fig. 60.

The safety cam shown in Fig. 59 is an important part of a Corliss gear. If for any reason the engine governor should fail to act, due, for instance, to the breaking of its driving belt, the governor would drop to its lowest position, supply more steam to the engine and allow it to run away. The *safety cam* prevents this by moving so far to the right that it strikes the hook when it descends to pick up the steam arm. The hook is consequently turned toward the right and then lifted without engaging the stud die; the valve conse-

quently remains closed and the engine stops.

Brown Releasing Gear. In addition to the *Reynolds hook*, several other devices are in use for opening and releasing Corliss admission valves. Among them the Brown releasing gear shown in Fig. 61 may be noted. The steam rod and dashpot rod are arranged much the same as in the Reynolds gear. The governor cam rod operates a plate cam having a curved slot so shaped that it takes the place of both the knock-off and the safety cam of Fig. 59. The steam arm is keyed to the valve spindle and carries at its lower end a steel die which is free to slip up and down a small amount. The part of this gear corresponding to the Reynolds bell crank becomes a straight rocker pivoted at its middle;

and the part corresponding to the Reynolds hook has at one end a die which engages the die of the steam arm, and at its other end a roller running in the curved cam slot. This hook is really a bell-crank lever with arms that are not in the same plane. The bearing on which it turns is carried on the lower end of the rocker, and therefore is equivalent to a movable pivot similar to the hook stud of the Reynolds gear.

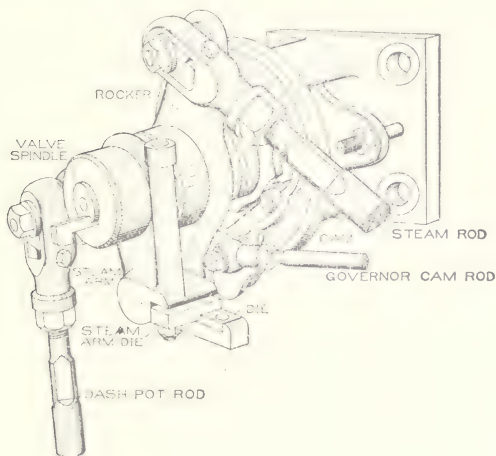


Fig. 61.

In the position shown the dies are engaged. Motion of the steam rod toward the right will move the lower end of the rocker toward the left, and consequently turn the valve spindle in a right-handed direction. This will open the valve and at the same time raise the dashpot rod. Meanwhile, the roller is moving toward the left in a circular part of the cam slot, the center of which is at the center of the valve spindle. This causes the steam arm and the bell-crank lever, which has the roller at one end, to move in such a way that there is no relative motion between them. As soon, however, as the roller comes to the point where it is forced to move out of this circular path and move farther from the valve spindle, both arms of the bell-crank lever are turned downward,

the dies become disengaged, and the dashpot closes the valve. The slight up-and-down motion of the steam-arm die allows it to rise while the hook die passes underneath when returning to re-engage for the next stroke. The makers claim that this gear permits a much higher speed than is possible with other Corliss gears.

Greene Gear. Another well-known drop cut-off gear is the Greene, shown in Fig. 62. The valves are of the gridiron type, sliding on horizontal seats, the admission valves parallel to, and the exhaust valves at right angles to the axis of the cylinder and just below it. AA are rock shafts turning in fixed bearings.

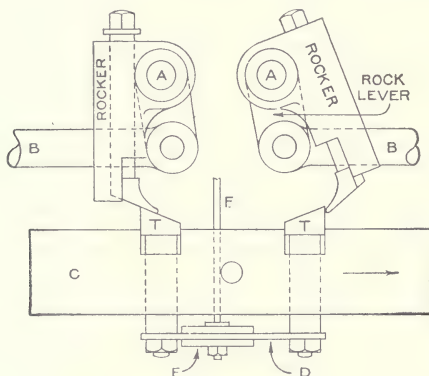


Fig. 62.

BB are the admission valve stems. C is a slide bar, receiving a reciprocating motion from an eccentric. TT are tappets connected to the slide bar. They move to and fro with the slide bar and can also move independently up and down. They are made fast at their lower end to the gauge plate D which slides through the guide E. The guide E is made fast to the governor rod F and through this means can be raised or lowered, thus regulating the height of the tappets.

As the slide bar moves toward the right, the right-hand tappet is brought into contact with the toe of the rocker, causing it to turn on its bearings and move the rock lever and the valve stem B toward the right, thus opening the admission valve. Since

the tappet moves in a horizontal direction while the toe of the rocker moves in an arc, it will be seen at once that they will soon become disengaged and release the valve which is at once closed by a dashpot not shown in the figure. If the governor raises the tappets, cut-off will be later. A nut at the bottom of the governor rod allows a proper adjustment of the guide and gauge plate. As the slide plate C moves toward the right, the left-hand tappet comes in contact with the heel of the left-hand rocker, both being beveled, it rises in its socket allowing the tappet to pass under. It then falls by its own weight and is ready to engage the tappet on its

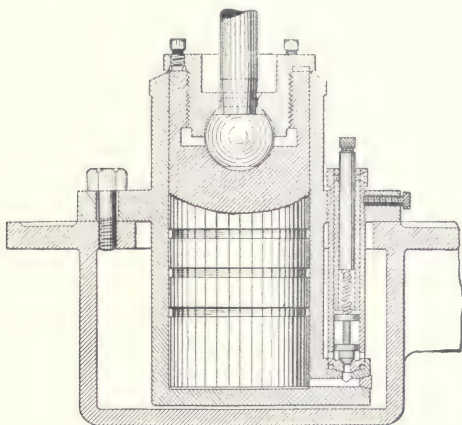


Fig. 63.

return and open the valve. In this gear the disengagement of the valve throws no load whatever on the governor which is a distinct advantage over the Corliss gear. The action of the exhaust valves is not shown in the cut.

The Sulzer Gear is a drop cut-off widely used in Europe. The valves are of the poppet type, lifting straight from conical seats, so that there is no friction. They are usually placed vertically above and below the cylinder axis and are operated by eccentrics from a shaft geared to the main shaft. The admission valves are lifted from their seats by suitable levers, then released by a

device equivalent in action to the Reynolds hook and are quickly closed by the action of springs.

The exhaust valves of all drop cut-off gears are comparatively simple in their operation and both in opening and closing they are moved by the direct action of the exhaust rods.

A common form of vacuum dashpot for closing admission valves is shown in Fig. 63. The rod coming down from the steam arm makes a ball-and-socket joint with the dashpot piston. The dashpot is often let down into the engine frame as shown. When lifted, the piston produces a partial vacuum underneath it so that it tends to drop quickly as soon as the valve gear is released. On some of the largest modern engines where the valves are very heavy, steam-loaded dashpots are used; that is, the dashpot piston has steam pressure on one side, and an air cushion on the other prevents it from striking the bottom of the dashpot.

Corliss Valve Setting. The setting of a Corliss valve gear

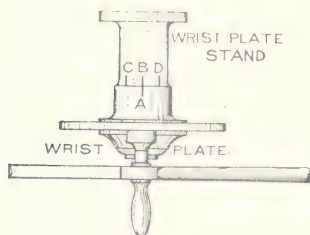


Fig 64.

is a much longer process than the setting of a plain slide valve, but is nevertheless a comparatively simple matter, for the various adjustments are nearly all independent of one another. In gears like that shown in Fig. 58 the length of both the eccentric rod and carrier rod are unusually adjustable, and the former should be of such length

that the carrier arm swings equal distances on each side of a vertical line through its pivot, and the carrier rod should be adjusted until the wrist plate oscillates symmetrically about a vertical line through its pivot. Nearly all Corliss engines have one mark on the wrist plate hub and three on the wrist plate stand, as shown in Fig. 64, and the wrist plate should swing so that A, the mark it carries, moves from C to D, but not beyond either one. When A is in line with B, the wrist plate is in mid-position. The valves are then not in their exact mid-position; but it is customary to regard them as being in mid-position, and to speak of the lap as the amount the valve covers the port when the wrist plate is in mid-position.

To set the valves, remove the bonnets or covers of the valve chambers on the side opposite the gear. The ends of the valves are circular, but inside their cross-section is as shown in Fig. 65. On the end, in line with the finished edge of the valve, and on the seat in line with the edge of the steam port, are marks as shown in Fig. 65. When these marks coincide, the valve is either just opening or just closing, and when in any other position, the amount of opening or the amount by which the port is closed is shown directly by the distance between the marks. Block the wrist plate in mid-position, hook up the admission valves and adjust the length of the steam rods by means of the right and left threads provided for the purpose, until the ports are covered by the amount of lap indicated in the following table opposite the given size of engine.

Dia. of Cyl. in inches.	Steam Lap in inches.	Exhaust Clearance in inches.
12	$\frac{1}{4}$	$\frac{1}{32}$
14 to 16	$\frac{5}{16}$	$\frac{1}{32}$
18 to 22	$\frac{3}{8}$	$\frac{1}{16}$
24 to 28	$\frac{7}{16}$	$\frac{3}{32}$
30 to 36	$\frac{1}{2}$	$\frac{1}{8}$
36 to 42	$\frac{5}{8}$	$\frac{5}{32}$

Next adjust the exhaust rods until the exhaust ports are open an amount equal to the clearance given in the above table. Set the engine on its head-end dead point, hook the carrier rod onto the wrist plate and in the direction in which the engine is to run, turn the eccentric enough to open the head-end admission valve by a proper amount of lead; then the eccentric will be 90° plus the angular advance ahead of the crank. The

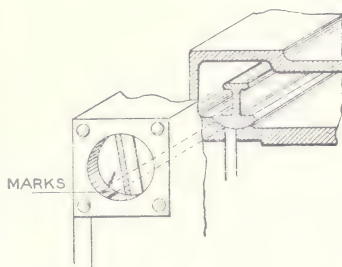


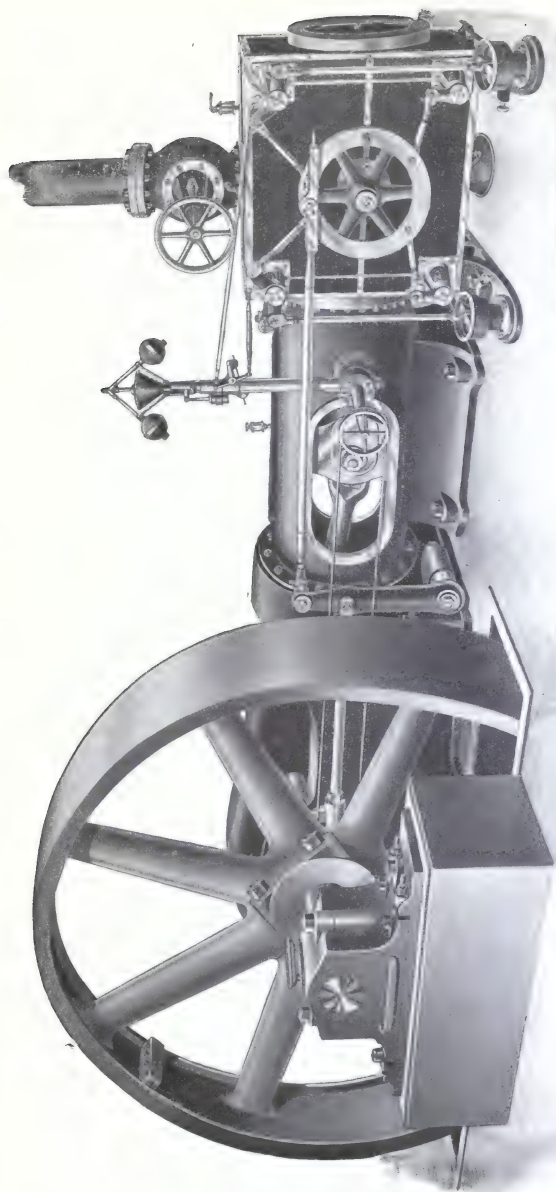
Fig. 65.

proper amount of lead will depend upon both the design of the gear and the speed at which the engine is to run; and may vary from $\frac{1}{32}$ " for small engines to as much as $\frac{5}{32}$ " or $\frac{3}{16}$ " for large and higher-

speed engines. When the proper amount of lead has been obtained, fasten the eccentric on the shaft by means of the set screw and make sure by trial that the wrist plate moves to its extremes of travel. The dashpot rods must be adjusted so that when the dashpot piston is at its lowest position, the hooks (see Fig. 59) descend just far enough for the hook dies to snap over the stud dies with about $\frac{1}{32}$ " to $\frac{1}{16}$ " to spare, depending on the size of the gear.

To adjust and equalize the cut-off, lift the governor to about the position that it will occupy when running at normal speed, and put a block under the collar to hold it in this position. First set the double lever at the right of the governor cam rods so that it makes approximately equal angles with each rod, and then turn the engine over by hand until the piston has moved to the desired point of cut-off. Adjust the proper cam rod until the knock-off cam strikes the hook and allows the valve to close, then turn the engine to the point of cut-off on the other stroke and adjust the other cam rod in a similar manner. Now set the governor in the lowest position to which it could fall if there were no load on the engine, and set the safety cams so that in this position the hook cannot open the valve. A latch is provided on which the governor can be supported slightly above its lowest position so that the valves can be opened by the hook when starting the engine. As soon as the engine speeds up this latch must be moved aside, so that if the governor fails to act, it can drop to its lowest point, otherwise this latch would hold it just high enough so that the safety cams could not act.

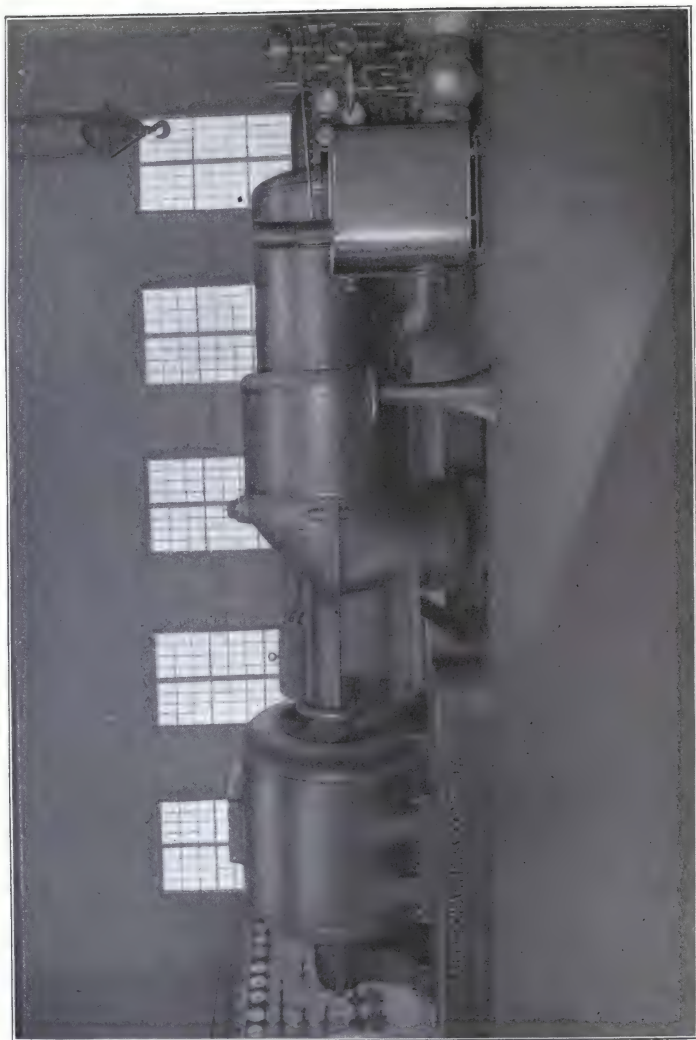
When Corliss gears are set as here described, the position of the eccentric may not be quite right, due to an incorrect estimate of the amount of lead required. The error is likely to produce faulty release and compression as well as poor admission, but it cannot be very serious, and the engine will turn over with its own steam, so that indicator diagrams may be taken. The final adjustments can then be determined from an examination of the diagrams.



FULTON IRON WORKS, FULTON, N.Y.

29

SINGLE CYLINDER HEAVY DUTY CORLISS ENGINE.
Fulton Iron Works.



STEAM TURBINE INSTALLATION IN A CENTRAL POWER AND LIGHTING STATION OF ONE OF THE PRINCIPAL
COAL PRODUCING DISTRICTS
Allis-Chalmers Co., Milwaukee, Wis.

STEAM TURBINES

PART I

Introduction. The steam turbine is one of the most recent engineering developments, and perhaps the most talked of, at the present time. During the past ten years the most marked improvements in its development have been made, and this has given a great impetus to engineering, especially steam engineering, although the very high speeds of rotation have driven the electrical engineer to work out new ideas in designing generators suitable for these higher speeds. The turbine has forged rapidly to the front, and, in spite of an early and serious handicap in the way of steam economy, has taken its place beside the best reciprocating engines of the present time. Many claim it to be superior in the matter of steam economy, but this will be discussed more fully later on. The turbine evidently possesses many advantages over the reciprocating engine, and, in its field of greatest usefulness, is likely to find in the near future a more severe competitor in the gas engine than in the reciprocating engine. For some classes of work, the steam turbine in its present state of development is entirely unadapted.

The steam turbine consists essentially of nozzles or guide passages which direct the steam onto vanes or buckets attached to the periphery of rotating wheels, the essential elements of which are shown in Fig. 1. The simplest form of turbine is perhaps one of the type in which a jet of steam impinges upon the buckets of a wheel, in much the same manner that a stream of water impinges upon the buckets of a Pelton water-wheel; there is, in fact, a great similarity between water turbines and steam turbines. The underlying principles are the same in either case, but the application of those principles is different. Steam flowing through a properly designed nozzle, with 150 pounds boiler pressure on one side, and the usual turbine vacuum on the other, will attain a velocity of about 4,000 feet per second, or about twice the muzzle velocity of a rifle

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ball. Water, to attain this enormous velocity, would have to flow from a head of about 234,000 feet. When this is compared with the ordinary water head of 150 feet or less, or even with such an exceptionally high head as 3,000 feet, which is sometimes met with in water powers on the Pacific Slope, a glimpse will be had of the magnitude of the problem confronting the steam turbine engineer. To put this in other words, the steam turbine designer has to deal with a velocity equivalent to that produced by a head of water nearly

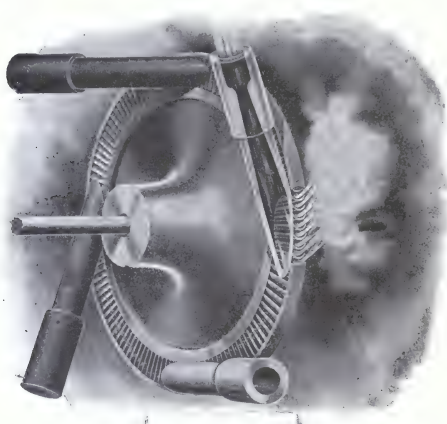


Fig. 1. Elements of DeLaval Turbine.

1,500 times as high as Niagara Falls. It will at once be seen, then, that the velocity of rotation of a simple turbine wheel to attain the best efficiency must be enormous. If this total head is to be used in one wheel, the peripheral speed must be nearly 2,000 feet per second, and at such speeds, the centrifugal force is so great that it is no easy matter to design a wheel that will not burst, even were there available some material stronger than any we now know. As it is, about 1,200 feet per second is considered the practical limit of peripheral velocity for a wheel built of the best nickel steel.

A little mathematical calculation will show that wheels of five feet in diameter will revolve 4,600 times per minute to attain a velocity of even 1,200 feet per second at the periphery. It is the problem

of the steam turbine designer to reduce these speeds to more manageable rates without at the same time making too great a sacrifice of efficiency.

From a thermodynamic standpoint, the turbine and reciprocating engine are not unlike, but the force of the steam acts differently in them. In both, it is the heat energy of the steam that does the work. In the one, the steam slowly expands, exerting pressure on a piston; in the other, it expands in narrow passages, pushing the particles ahead faster and faster and thus obtaining velocity which is then imparted to the vanes of a rotating wheel. In the one, the steam acts by virtue of statical pressure; in the other, by virtue of its high velocity. In either case, it is the internal heat of the steam that causes the expansion and does the work. If heat is lost in any way, by condensation, radiation, etc., the work will be proportionally less. In the turbine, a difference in pressure from inlet to outlet acts as a motive force indirectly, and then only in so far as it causes a rapid flow of steam.

Advantages. The well-known expression, $work = force \times space$, embodies the idea that a given amount of work may be accomplished in a certain time by increasing the total force of the steam on the piston at the expense of the number of revolutions of the fly-wheel per minute, or *vice versa*. For example, a Corliss engine running at 180 revolutions per minute requires a mighty effort behind the piston to develop 1,000 horse-power, and this tremendous force demands a large cylinder, a heavy frame, and an immense fly-wheel. If, now, an engine were built to run at 800 revolutions per minute, much less push behind the piston would be necessary to develop the same horse-power, and, therefore, the parts could all be made smaller, and the whole weight very much reduced.

To go a step further, and consider the steam turbine, which must run at 2,000 to 3,000 revolutions per minute, it is clearly seen that this enormous speed reduces the mass of the parts even more. The heavy fly-wheel is no longer necessary, as the rotating parts are moving at a sufficiently high speed to acquire an immense inertia, and there is always a constant effort exerted on the vanes by the team, thereby producing an absolutely steady turning moment. Furthermore, the motive parts of the turbine revolve, which is in direct contrast to the reciprocating engine, in which the piston is

moving backward and forward, and the turning moment is continually changing from a maximum to a minimum. It is clear, therefore, that for a given horse-power, the steam turbines produce smaller machines, lighter foundations, and consequently smaller power houses. A fair idea of the relative space occupied may be gained from Fig. 2.

Again, the generator, on account of this high speed, will be smaller and less expensive. The turbine requires oil in its bearings

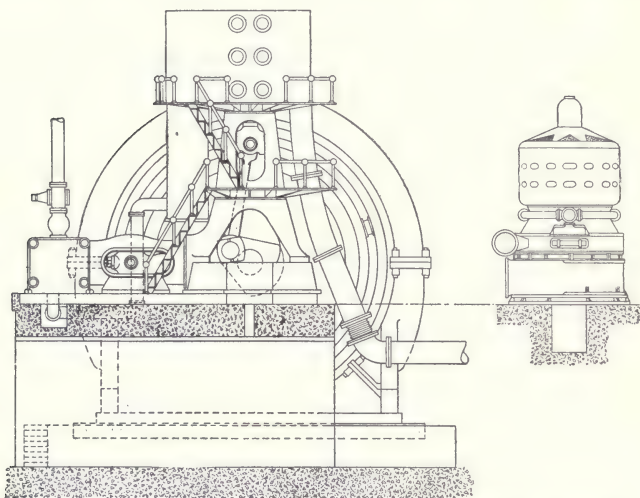


Fig. 2. Comparative Sizes of 5000-K. W. Corliss Engine and Generator and Curtis Turbo-Generator of Same Power.

only; hence there is no oil to go over in the condensed steam, and the condensation may be used for boiler feed without any danger of carrying oil into the boilers. The turbine requires somewhat less attendance than the reciprocating engine, and the whole machine is compact and simple. To do its best, the turbine requires a higher vacuum than is ordinarily obtained for the reciprocating engine, and hence needs very much larger condensers, more cooling water, and additional air-pump capacity. All this in a measure offsets some

of its advantages, and frequently more trouble arises from the air pumps and condensers than from the turbine itself. The turbine may of course operate at the usual vacuum with a somewhat greater steam consumption and a slightly lower efficiency.

The reciprocating engine has its own advantages, and in certain classes of work will doubtless hold its own, but for all such apparatus as blowers, centrifugal pumps, generators, etc., which may be direct connected to a turbine, the reciprocating engine is rapidly becoming a thing of the past, and even for factories where belt drives are used, the steam turbine has been suggested.

History. The steam turbine is not only one of the most recent engineering developments, but is, at the same time, perhaps, one of the most ancient forms of prime mover. In a book written by Hero of Alexandria, over 100 years before the beginning of the Christian era, a very simple form of steam turbine is described. It consisted of a hollow sphere mounted on hollow trunions, through which the steam passed into the sphere. On opposite sides of the sphere were outlets consisting of pipes bent at right angles in lines tangent to the equator

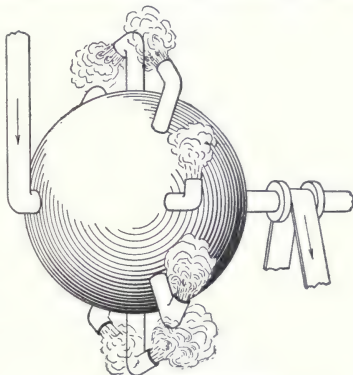


Fig. 3. Hero's Steam Turbine.

of the sphere, in such a manner that the reaction of steam escaping through these pipes caused the sphere to revolve on its trunions, in much the same way that the water escaping from the arms of a lawn sprinkler causes it to revolve. This turbine, which is illustrated in Fig. 3, is the simplest form of the pure reaction motor.

In 1629, Branca, an Italian, invented a turbine much like a miniature water wheel, which was driven by a jet of steam from a nozzle directed against the buckets of the wheel. This is the simplest form of an impulse turbine, and is illustrated by Fig. 4.

In 1784, Wolfgang de Kempelen designed a turbine of the lawn sprinkler type, similar in principle to Hero's engine, the chief dif-

ference being the substitution of a horizontal revolving tube for the hollow sphere which Hero used. Steam, escaping from the outlets in opposite ends of the tube, caused it to revolve by reaction, just as the escaping water causes the lawn sprinkler to revolve.

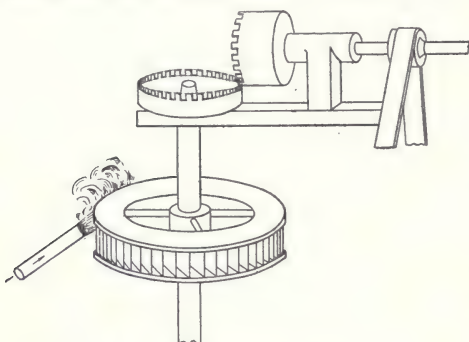


Fig. 4. Branca's Impulse Turbine.

In 1784 it is said that James Watt took out a patent on a turbine, but, as we all so well know, devoted his genius to the development of the reciprocating engine. At this time, both types of engine were in about the

same crude form, and it is possible that had Watt devoted his energies to the turbine instead of to the reciprocating engine we might not have had the ordinary form of steam engine in its present

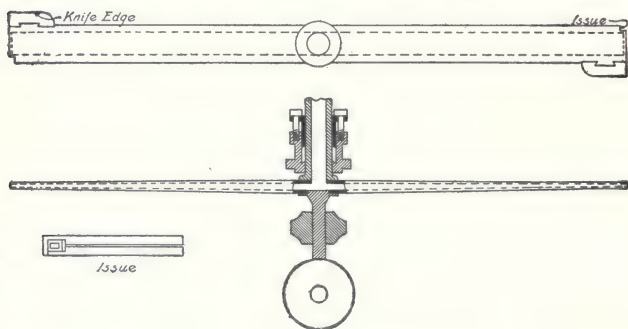


Fig. 5. Reaction Wheel of Avery & Foster.

high state of efficiency, for the turbine possesses so many advantages. This is proved by the fact that it has at last come to the front in spite of the great commercial success of the reciprocating engine.

In 1831, Avery & Foster took out the first patent granted for a turbine by the United States Patent Office. This was on the Hero lines, and was really an improvement on the Wolfgang de Kempelen turbine of 1784. This turbine appears to be the first to attain commercial success. Several were built under the Avery patent and were used to run sawmills near Syracuse, N. Y.

Steam entered a hollow shaft, Fig. 5, through a stuffing box, passed through to the hollow arms, and escaped through plain openings in opposite ends of the arms. The speed of rotation was enormous, the periphery of a 7 ft. wheel traveling at the rate of about 14 miles per minute. The wear was excessive, and this, combined with inability to get proper packing for the stuffing boxes, rather than the lack of steam economy, doubtless caused its failure, for the reciprocating engine of those days had not reached its highest state of economy. Had Avery used the present expanding nozzles instead of plain openings for his steam outlets, his steam consumption would doubtless have been less, but the speed of rotation more. Diverging nozzles were used as early as 1838, but as they were not correctly proportioned, they were a hindrance rather than a help, and the idea seems to have been given up for a time.

As early as 1827, a compound turbine was patented by Real & Pichon, the idea being to reduce the velocity of rotation by passing

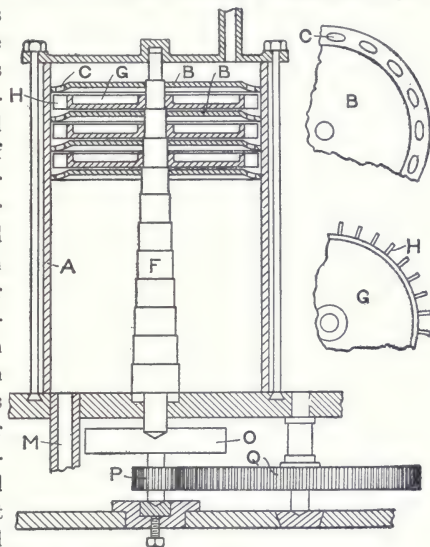


Fig. 6. Real & Pichon Compound Turbine.

the steam through successive wheels *G*, Fig. 6, separated by disks *B B* containing outlets *C* to permit the passage of the steam from one chamber to another. *H* is one of the blades, *F* the shaft, and *M* the steam exhaust. This is the principle on which the present Rateau turbine works.

The chief cause for early failure in turbine work was lack of comprehensive knowledge of the flow of steam. It was not until 1840 or thereabouts that anyone seemed to get at the real facts or appreciate the true significance of the situation. In this year Pilbrow patented a machine that was a distinct advance in the right direction, and his patent claims show that he, at least, understood some of the fundamental principles. In 1842, he attempted to reduce the speed of rotation by compounding, passing steam through

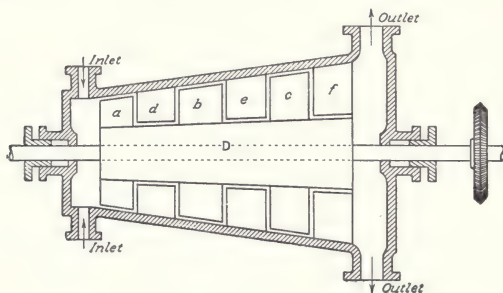


Fig. 7. Wilson's Compound Turbine.

successive wheels revolving in opposite directions. There are, of course, grave objections to such an arrangement. He later invented a turbine with several nozzles that could be successively cut out of action as the load on the turbine varied. This, in its crude form, is the fundamental idea of the arrangement of the nozzles used in the DeLaval and Curtis turbines, but Pilbrow used a converging, instead of a diverging nozzle, and his wheel was unlike others of the impulse type.

In 1840, Wilson patented the forerunner of the Parsons type of turbine. He passed steam successively through rows of running and stationary vanes, gradually expanding it until the exhaust pressure was reached. A view of Wilson's invention is shown in Fig. 7; *a*, *b*,

and *c*, are vanes which are attached to and rotate with the drum *D*, while *d*, *e*, and *f* are stationary guide vanes. Steam enters at the left, passes through the turbine longitudinally, and exhausts at the right. Wilson appears to have been among the first to realize that the volume of steam increases as it expands to lower pressures, to provide for the same by increased size of passages, and, what is perhaps most important, to claim this in his patent.

In 1858, Hartman Bros. patented a turbine consisting of two revolving disks *c* and *c'* fixed to a shaft *D*, as shown in Fig. 8. Between them was a segment of stationary reversing blades *d d*. Steam entered from a nozzle *F* and was exhausted at *H*; *G* is the casing. This turbine embodied the essential element of the one-stage Curtis turbine of the present day.

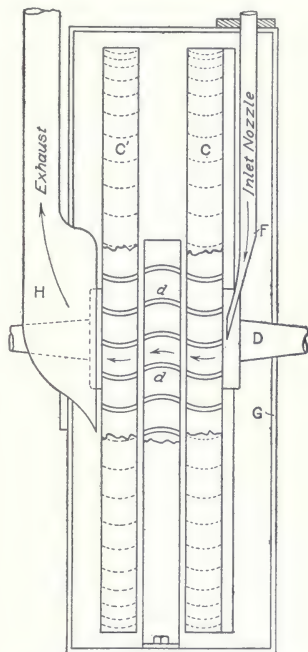


Fig. 8. Hartman's Compound Impulse Turbine.

Perrigault & Farcot, about 1870, patented a compound turbine in which the steam, as it left the buckets, passed through successive passages and again and again impinged upon the face of the same wheel. This is the principle adopted in the Riedler-Stumpf turbine and is illustrated by Fig. 9. Steam enters this turbine through the nozzle *A*, passes through the wheel buckets to the other side, and discharges into pipe *B*, which brings the steam around again to the inlet side. Here it discharges through the opening *b*, against another bucket of the same wheel, whence it is picked up by the opening *x*, in the pipe *C*, and so on, finally exhausting through the pipe *D*. Somewhat earlier, a turbine was invented that returned the steam in a similar

manner, but to another set of blades on the same wheel. This idea has also been perfected by Riedler & Stumpf.

Moorehouse patented, in 1877, an improvement on the type suggested by Real & Pichon in 1827. His chief claim was an allowance for the increased volume of expanding steam in this type of turbine.

DeLaval took out a patent on a reaction turbine of the Hero type in 1883. It differed from the Avery turbine in detail, but not in principle. This turbine was extensively used for running cream separators, and was commercially successful, but was later abandoned for the present type of DeLaval motor.

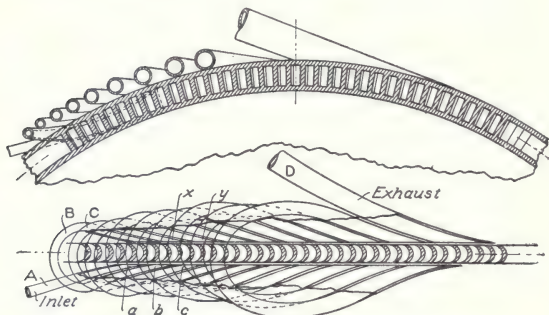


Fig. 9. Compound Turbine with One Wheel.

In 1885, Parsons took out his first turbine patent on a motor along the lines previously suggested by Wilson, and is responsible for the successful development of this type of motor. His first turbine, shown in Fig. 10, took steam in the center *A*, and exhausted at both ends through the exhaust passage *EE*, thus avoiding any end-thrust on the shaft *B*. At the same time, he patented his famous flexible bearing, now in general use. In 1888, he patented the present arrangement of grouping several rows of blades together increasing the drum diameters step by step to provide for proper expansion, at the same time patenting his balancing pistons, at present employed to relieve end-thrust.

The expanding nozzle had been patented in 1867 for use in steam injectors, but it was not until 1894 that anyone patented its use in

connection with a turbine. In this year, DeLaval secured this patent and used the nozzle in connection with his turbine, for the purpose of expanding the steam and getting a high velocity of jet with increased kinetic energy.

During 1894 and 1895 there were issued a large number of patents, many of which have been successfully developed. Among them were Parsons', Rateau's, and the first patents for the use of buckets of the Pelton type.

In 1896, Curtis patented the use of an expanding nozzle in combination with a compound wheel of the type suggested by the Hart-

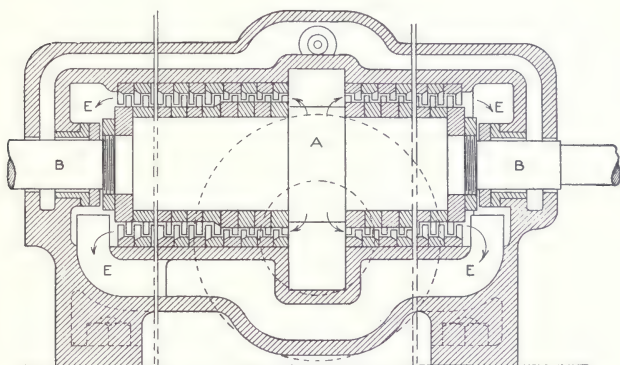


Fig. 10. Early Parsons Turbine.

man patents in 1858. Others had used both the expanding nozzle and the same type of wheel, and only two years earlier patents were taken out for a converging nozzle with a similar wheel. From a study of nozzles, it will appear that the converging nozzle could be used economically by increasing the number of stages used in the expansion, but the turbine would be larger than the Curtis, and probably less efficient.

Patents were issued in 1898 to Riedler & Stumpf, whose turbine appears to be an improvement on the Perrigault & Farcot, patented about 1870.

In 1900, the Zoelly patents were issued. This turbine in principle is similar to the Rateau, but different in construction.

The steam turbine patents issued since 1900 are altogether too numerous even to mention, but from them a number of commercial machines have been developed, and are now on the market. The principal commercial turbines will be described later.

It must not be thought that this summary is at all exhaustive, or that even all the noteworthy turbine patents have been mentioned. There are hundreds of them, and it is possible here only to mention those that are the immediate forerunners of our present commercial types. This brief summary will show that the commercial success of the turbine has been due to a more complete knowledge of the properties of steam, improved details, and the possibility of better workmanship, rather than to the development of new principles, for the distinctive fundamental ideas of all of our commercial turbines had been suggested years ago. It should be borne in mind, however, that no fundamental principle can be successfully worked out unless the minutest detail is correct, and these details may, and in the case of the turbine did, prevent the successful carrying out of the early ideas.

Fundamental Principles. The underlying principles of steam and water turbines are alike—a moving fluid impinges upon curved vanes or buckets attached to the periphery of a wheel; thus causing it to revolve. The vanes or buckets change the direction of the actuating fluid and absorb part of its energy, the fluid leaving the turbine with a comparatively low velocity. To insure reasonable economy, the fluid must impinge upon the vanes in a direction tangential to their surface at the point of impact, so as not to impart any shock and to avoid spattering. Further, the residual velocity at outlet should be as low as possible.

In either class of turbine, rotation is caused, not by the statical pressure of the actuating fluid, but by the velocity which it imparts to the rotating turbine wheels. The kinetic energy of the fluid passing through the turbine is equal to $\frac{WV^2}{2g}$, where W equals the weight of the fluid per second and V is its velocity at entrance. Evidently, the smaller W , the larger V must be to develop the same power. If the fluid leaves the turbine with the velocity V_a , then $\frac{WV_a^2}{2g}$

represents the energy not absorbed by the turbine. If V_a is small, this wasted energy will be likewise small.

Since the fundamental principles of the turbines are the same, it would seem at first sight as if steam and water turbines could be built on similar lines. But this is not so, because the difference in density and in elasticity of the two fluids requires different applications of those principles. In the steam turbine, not only must proper steps be taken to abstract the energy from the steam jet, but also to make that energy a maximum by providing for the proper expansion of the steam.

To make more clear the differences just mentioned, it should be remembered that water is an *inelastic* fluid; that is, one having a constant volume under all conditions of pressure. Therefore, in flowing through a nozzle, if the velocity at the outlet is to be greater than the velocity in the pipe, the area of the outlet must be smaller than the cross-section of the pipe. Steam, on the other hand, is an *elastic* fluid and expands rapidly as it flows through a nozzle. If the increase in volume were in exact ratio to the increase in velocity, then, for maximum efficiency, the nozzle would be parallel-sided. This happens when the pressure at discharge is about 60% of the initial pressure. But when discharging into a low pressure, the volume of the steam increases more rapidly than the velocity, and hence, if the mouth of the nozzle is to be capable of discharging the same weight of steam per second as the throat (the condition for maximum efficiency), the cross-sectional area of the nozzle must constantly increase toward the outlet.

Steam will expand as it passes through the turbine and if the passages are correctly proportioned, so that this expansion can take place only in one direction, that is, in the line of flow, the steam particles will be forced forward in a nearly uniform jet; the steam, by virtue of this expansion, will attain a very high velocity and the jet will consequently have a high kinetic energy.

Water turbines use a relatively small head and a large quantity of fluid; with the steam turbines, the quantity of fluid is small, but the head is very large. To develop large powers with any form of turbine, it is necessary that a number of wheels be used. With water turbines, each wheel acts under full head, each using a relatively small quantity of water. With steam turbines, however, it is the

head that must be divided into different steps; *i. e.*, a single steam turbine can use the full quantity of fluid but, if desired to run at relatively low speeds, it can use but a portion of the total head. To develop 1,000 H. P. on a turbine shaft, with a head of 150 feet, would require approximately 4,800 pounds of fluid per second, depending somewhat upon the design of the wheel. With a head of 3,000 feet there would be required 242 pounds of fluid per second, and with a head of 234,000 feet, comparable with that of a steam turbine, the requirement would be about $3\frac{1}{2}$ pounds of fluid per second. It is thus clearly seen that the difficulty in developing large powers with the water turbine is that of providing

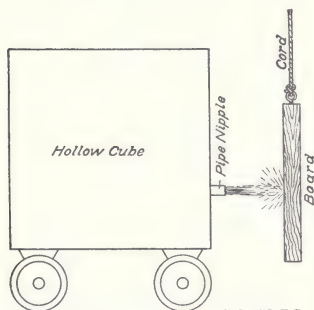
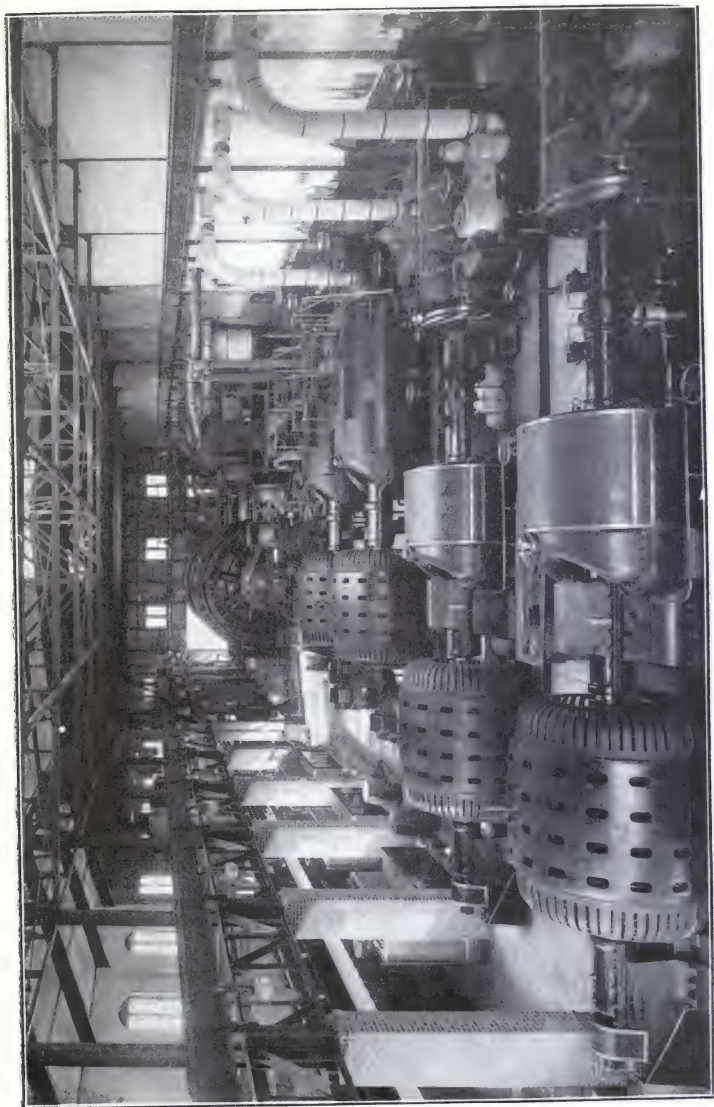


Fig. 11. Apparatus for Showing Force of a Jet.

for a sufficiently large quantity of fluid through the turbines, and with the steam turbine, that of handling the great velocities resulting from the enormous head.

In all steam turbines, the steam is expanded in suitable nozzles or passages. In some the expansion is all in the nozzles; in others, partly in the nozzles and partly in the vanes or blades. In some, the total expansion from boiler to exhaust takes place in one nozzle, and the energy is absorbed by a single wheel; such a turbine is called a *single-stage turbine*. In others, the expansion in one set of nozzles is only partial, and after passing through one or more wheels, the steam again passes through another set of nozzles and set of wheels, and so on, until exhaust pressure is reached. This is a *multi-stage turbine*. There may be all the way from one to forty stages, or even more, in turbines of this type where all expansion is in the stationary vanes; and in turbines where part of the expansion is in the running vanes, there may be 100 stages or more. Some turbines use nozzles for expanding the steam, and some use stationary vanes for the purpose, these vanes being so shaped as to provide suitable passage areas to permit of steam expansion. The principle is the same whether nozzles or blades are used, but blades are generally



GROUP OF WESTINGHOUSE-PARSONS TURBINE UNITS IN THE 16,000 H. P. PLANT OF THE PHILADELPHIA RAPID TRANSIT CO.
The Westinghouse Machine Co.

used where many stages are employed and the drop in pressure is small from stage to stage.

Before taking up the actual study of steam turbines, it will be necessary to have a clear conception of a few elementary principles of mechanics. Suppose a hollow cube to be filled with some fluid (water or steam) at a given pressure, and to have an opening in one side that can readily be closed. The arrangement is such that when the outlet is opened, the internal pressure will remain the same. If the outlet is opened, the fluid will rush out, as shown in Fig. 11, and, if the jet is supposed to strike against a board free to move, the jet will exert a force upon that board tending to swing it in the direction of the jet. This force is called an *impulse*. At the same time there will be a tendency on the part of the cube to move in the opposite direction, and the force thus developed is called a *reaction*. It may be explained in this way:

Suppose each side of the cube to be one foot square, the area of the opening, one square inch, and the internal pressure, 100 pounds per square inch. There will be 144×100 pounds pressure on each side of the cube with the outlet closed, but when the one-inch outlet is opened, the total pressure on the side containing the outlet will be reduced by the pressure of 100 pounds on the opening itself. This will leave an unbalanced force of 100 pounds acting in the opposite direction, which is the origin of the reactive force. This explanation is not strictly correct, but serves to give an idea of these two forces, impulse and reaction.

Hero's turbine was a reaction turbine pure and simple, Branca's, an impulse turbine; but what is called a reaction turbine at the present time is not a simple reaction turbine in any sense, but one running under the combined influence of reaction and impulse. Likewise, the so-called impulse turbine is not a pure impulse turbine, but acts under the combined influence of impulse and reaction. There is no pure reaction turbine now on the market. The so-called impulse turbine being rather simpler of explanation, for the present only this type will be considered in the following explanations. How

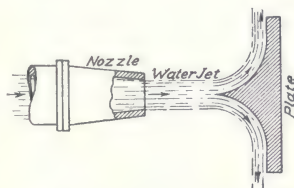


Fig. 12. Jet Deflected through 90°.

these principles apply to the so-called reaction turbine will be explained later.

Suppose a stream of water from a nozzle to impinge upon the plate shown in Fig. 12, and so made that the jet is divided, and without shock departs in a direction tangential to the plate and at 90° to the line of impact. If the velocity of impact of the jet is V feet per second, its velocity in the same direction after striking the plate will be zero, and therefore, a definite force will be exerted on that plate, equal to the force necessary to impart a velocity of V feet in one second to the mass of water in the jet. The acceleration, therefore, will be V feet per second, and since force is measured by mass times acceleration, this force, acting on the plate, will be $F = MV$. If the plate is allowed to move in the direction of the jet with a

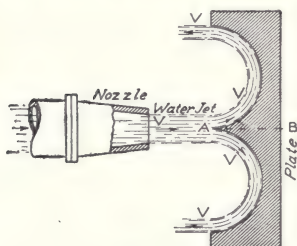


Fig. 13. Jet Deflected through 180° .

velocity V_1 , the relative velocity of the plate with reference to the jet will be $V - V_1$, and the corresponding force acting on the plate will be $F = M(V - V_1)$. Since work is measured by the product of force and distance, the force acting through the space V_1 in one second, will do the work $W = FV_1 = M(V - V_1)V_1$ foot pounds.

Now if the plate were shaped as shown in Fig. 13, so that the direction of the jet were completely reversed, that is, turned through 180° , there would be an additional pressure on the plate, due to the reaction of the jet leaving it. This, neglecting friction, would be equal to the original impulse, thus making the total force on the plate $2F$ instead of F . It is quite evident that if the force is twice as great, the work must also be double, and the above expression for the work done becomes

$$W = 2FV_1 = M \times 2(V - V_1)V_1$$

For this reason, turbine vanes are made so as to reverse the direction of the jet as completely as possible. Complete reversal is not practicable because some clearance must be allowed for the deflected jet to escape. This is especially true in the usual case in practice, where the jet impinges upon the vanes from the side. Here, the angle has

to be such that the revolving wheel will clear both nozzle and deflected jet.

If the bucket shown in Fig. 13 were held stationary, the force exerted by the jet would evidently be a maximum and equal to $2MV$; but the velocity of the bucket being zero, the work, equal to the force multiplied by the space, would also be zero. If, on the other hand, the velocity of the bucket were equal to the velocity of the jet, the push would be zero, and the work again zero. Somewhere between these limits, there must evidently be a velocity which will produce maximum results.

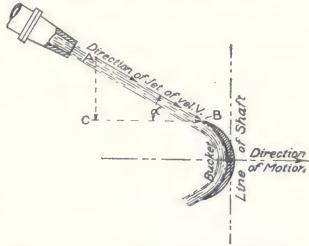


Fig. 14. Jet Impinging upon Curved Vane at an Angle with Plane of Rotation.

Suppose now, that $V_1 = \frac{V}{2}$:

then

$$W = M \times 2 \left(V - \frac{V}{2} \right) \frac{V}{2} = M \frac{V^2}{2},$$

but

$\frac{M V^2}{2}$ = the kinetic energy of the jet, as it issues from the

nozzle. Therefore, if the speed of the bucket is one-half the velocity of the jet, we have an efficiency of 100%, neglecting losses, and this is, of course, the best obtainable. Therefore, the greatest efficiency is obtained when the speed of the bucket is half the jet velocity, provided the jet impinges upon the bucket in a direction parallel to the line of movement of the bucket. For other angles, the speed for maximum efficiency would be somewhat less.

If a jet with the velocity V strikes the bucket at an angle α , as shown in Fig. 14, its velocity AB could be resolved into two components—one CB at right angles to the shaft, and one CA parallel to the shaft. The one at right angles to the shaft, commonly known as the *velocity of whirl*, would produce a rotative impulse equal to $V \cos \alpha$, and V_1 , the velocity of the vane necessary for maximum efficiency, would be half this, or $V_1 = \frac{1}{2} V \cos \alpha$, provided the angle with which the jet leaves the blade is equal to the angle of impingement. The component AC , parallel to the shaft, would have no

unless the jet angle α is zero, and the direction of the jet is reversed through 180° , an impracticable condition.

Nozzles. Steam does not cause rotation in the turbine because of its static pressure, but, as already stated, because of its velocity, a difference in pressure acting indirectly, by imparting velocity to the steam. It is evident, then, that in this class of motor, steam velocities are all-important. The steam possesses energy by virtue of the heat which it contains, but to make this energy available in the turbine, it must be transformed into kinetic energy by the production of a high jet velocity. The correct shaping of the nozzle is the all-important factor in acquiring the requisite steam velocity, as will appear from the following considerations:

It has been well established by experiment that steam at high pressure flowing into a space at lower pressure, through a nozzle with parallel sides, cannot attain a velocity exceeding 1,450 to 1,500 feet per second, no matter how high the initial pressure nor how low the pressure into which the steam discharges. This limiting velocity is due to the fact that at the throat of any nozzle there occurs a drop in the pressure of the steam to about 58% of the initial pressure, and, if the nozzle be a cylindrical one, this drop will remain practically constant throughout the length of the nozzle. The velocity acquired by virtue of this difference in pressure will therefore be about the same, whether the absolute pressure into which the steam is discharged is 58% of the initial pressure or much less. In the latter case, the throat pressure cannot change until the outlet is reached, when the pressure drops suddenly to the pressure of the space into which the steam is discharging, and the steam immediately expands in all directions, thus dissipating its energy. The only case in which maximum efficiency is developed with orifices and short passages with parallel sides is when the low pressure is greater than 58% of the high pressure.

This limiting value for orifices or parallel-sided nozzles, and the consequent limit of steam velocity, makes it impossible to develop the greatest energy of the steam when expanding to low pressures except through a nozzle with flaring sides, in which the outlet is greater than the inlet. In such a nozzle the steam expansion occurs gradually in its flow, and is constrained to take place only in the direction of the flow. In this way, the velocity of the steam par-

ticles is increased as it proceeds along its nozzle until a tremendous speed has been developed, which will produce about 95% of the available energy. Furthermore, steam may be expanded effectively within the confines of such a nozzle *from any high pressure to any lower pressure*, provided the increase of areas of cross-section of the nozzle is proportional to the increase of specific volumes of the steam.

In other words, with a cylindrical nozzle, a limiting steam velocity of 1,450 to 1,500 feet per second is possible, no matter whether the initial pressure of the steam be 70 pounds or 200 pounds, or whether the pressure into which the steam is finally expanded be 58% of the initial or a 28" vacuum. Of course, as the weight of the steam per cubic foot varies with the pressure, a greater *weight* of steam will be discharged per second at higher pressure, resulting in a somewhat greater kinetic energy in the steam jet. On the other hand, if the nozzle has flaring sides, the steam at, say, 150 pounds gauge pressure,

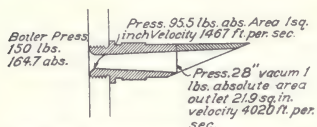


Fig. 16. Properly Designed Expanding Nozzle.

will have the same pressure at the throat, that is, 58% of the initial pressure, but will acquire a rapidly increasing velocity from throat to outlet, and, with a 28" vacuum ahead of it, will leave the nozzle with a velocity of 4,000 feet per second, assuming no friction in the nozzle. Fig. 16 shows a properly designed nozzle for expanding steam from 150 pounds boiler pressure to 28" vacuum.

Compounding. It has been explained that if steam is expanded in a suitable diverging nozzle, nearly all the heat energy becomes available as kinetic energy, and that this steam, when flowing from a boiler pressure of 150 pounds to a vacuum of 28", may attain a velocity of approximately 4,000 feet per second. If the linear velocity of the buckets were to be approximately one-half the velocity of the jet, there would be grave danger that the wheel would burst from centrifugal force. A peripheral speed of 1,200 feet per second is the limit in practice. 2,000 feet per second would mean about 12,750 revolutions per minute in a wheel 3 feet in diameter. This latter velocity would mean great delicacy in balancing and difficulty in providing suitable bearings, even if material could be found to withstand the strain. A wheel 15 feet in diameter would have to

revolve 2,500 revolutions per minute if the above mentioned peripheral velocity were to be obtained, and again the impossibility of construction is evident. Some means must therefore be employed to reduce the speed to manageable rates without unduly increasing the size of wheel. This may be done in a single-stage turbine by means of gearing, but here, if 1,200 feet is to be the maximum permissible peripheral velocity, and 2,000 is the theoretical velocity, there will be a loss of efficiency. As a matter of fact, the steam jet does not strike the wheel in a line at right angles to the shaft; consequently the velocity of whirl, as already seen, is $V \cos \alpha$, and with friction allowance, this is somewhat reduced, but, even with a bucket velocity of 1,200 feet per second, the revolutions will usually be too high.

Turbine speeds may be satisfactorily reduced without the use of gearing, by what is called *compounding*; i. e., by dividing expansion into separate stages, called *pressure compounding*; by passing the steam over several wheels with guide vanes between to redirect the steam upon the vanes in the next wheel, called *velocity compounding*; or by a combination of these two methods.

Suppose we start with steam at 150 pounds gauge pressure and expand it to 28 in. vacuum, not in one expanding nozzle but in several stages, so that the expansion in each would be to only about 60% of the next higher pressure, in which case, diverging or expanding nozzles would not be needed. The velocity of flow of the steam would be somewhat less than 1,450 feet per second at pressures above the atmosphere, and would decrease slightly as the pressure lowered; the lowest velocity, when discharging into a vacuum of 28 in., would be about 1,250 feet per second, but, by letting the drop in pressure be somewhat less than 60% in the higher stages, the velocity of flow could be made approximately 1,250 feet per second throughout. This is, of course, neglecting all losses. We could then have a steam speed of about 1,250 feet per second to deal with, instead of 4,000; the

peripheral speed of the buckets would be $\frac{1,250 \cos \alpha}{2} = 625 \cos \alpha$, or, when α , is small, about 600 feet per second. For a wheel 5 feet in diameter, this would mean about 2,300 revolutions per minute, and the conditions arising from such a speed are much more easily taken care of. This reduction in speed could be accomplished in about ten stages. To reduce the speed to half, or 300 feet per

second, would require four times as many stages because the number of stages would be equal to the square of the ratio of reduction of the steam velocity. Thus, the reduction from 4,000 feet per second to 1,250 feet per second will require $\left(\frac{4,000}{1,250}\right)^2 = 3.2^2 = 10$, approximately. To reduce the speed of the buckets from 600 to 300 feet per second, would evidently require four times as many, or 40 stages.

Suppose, now, the compounding were all in velocity stages, and that the expansion occurred in one nozzle. The velocity of steam would be nearly 4,000 feet per second, but it would have to pass through three sets of revolving wheels to bring the relative speed at each wheel to approximately 1,300 feet per second, and thus get the same speed of revolution as in the previous case.

This method of compounding gives a very compact form of turbine and one that has many mechanical advantages; but the wheels have to revolve in a bath of steam which makes the friction excessive, and the efficiency correspondingly lower. This was the idea of the original Curtis patent, but was soon abandoned for the combined pressure and velocity turbine which is to-day the principal feature of the Curtis design. In this combined method, there are two or more pressure stages, and in present practice, not over two velocity wheels and one set of guide vanes to each stage.* The older Curtis had even three or four revolving wheels per stage with a corresponding number of sets of guide vanes.

Compounding has been tried by the use of counter-running wheels, but with little success. If the guide vanes were on wheels free to turn, they would run in a direction opposite to the others, and if the relative peripheral velocity of the two were, say, 1,200 feet per second, it would mean that for each wheel the absolute velocity would be half this, or 600 feet per second. The difficulties of building and operating such a machine are considerable.

Types of Turbines. There are two main groups into which steam turbines are usually divided, one known as the *impulse*, and the other as the *reaction* type. It has become the general practice

*This does not apply to marine practice, the peculiar conditions of which warrant the use of a larger number of velocity wheels per stage. Small Curtis turbines and some special machines have three velocity stages.

to classify turbines under one or the other of these two general heads; but, as a matter of fact, every commercial turbine of the present time really develops its power under the combined influence of action and reaction. Yet there is a distinct difference between the expansion of steam in these two types, as, for instance, in the DeLaval and in the Parsons turbine. The use of the terms *impulse* and *reaction* in reference to turbines is undoubtedly unfortunate, but since their use has become practically universal, it is necessary to understand the significance of their application.

In the so-called impulse turbine, the steam, expanding in a nozzle or other suitable passage, thus attains a high velocity, and impinges upon the vanes of a rotating wheel. The steam, in passing through the wheel, gives up a part of its kinetic energy to the revolving vanes, and leaves the wheel at a lower velocity, but at the same pressure at which it left the nozzle. In the so-called reaction type, the steam enters the turbine at boiler pressure, passes through guide passages

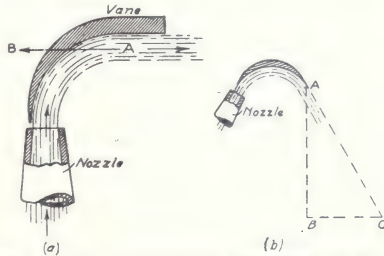


Fig. 17 Two Arrangements of Jet and Vane. (a) Pure Action. (b) Action and Reaction.

onto the vanes of a rotating wheel, and little by little expands as it passes through these vanes to subsequent guide passages and other vanes, the pressure gradually becoming lower; the velocity which is gained by the expansion in the guide passages and revolving vanes is practically all imparted to the rotating drum. The pressure is less on the one side of the vane than on the other, while, in the impulse type, the pressure is the same on both sides of the vane.

The engines of Hero, Wolfgang De Kempelen, and Avery were all purely reaction types, but the Parsons acts by impulse as well as reaction, and the Curtis and DeLaval, by reaction as well as by impulse. To make this clear, consider Fig. 17 (a), which shows a vane and jet. The vane is so shaped that the jet leaves it at right angles to the direction of impact. Here is a case of pure action, so far as any force tending to move the vane in a direction parallel to the

in the revolving vanes as well as in the guide passages, and the vanes and guides are placed so as to give a constantly increasing area of passage to allow for the increasing volume of the steam as it expands. As the steam expands in the guide passages, it acquires velocity and impinges upon the running vanes, thus giving a decided impulse to them, and as it again expands in the running vanes, the reaction produces a further impelling force.

The distinguishing feature, then, between these two distinct types of turbine is not to be found in the impulse or reaction of the steam at all, for both types, as we have seen, act by virtue of both forces; but the distinction lies in whether the expansion of the steam takes place fully in a set of nozzles or guide passages with no expansion in the moving vanes, or whether the steam expansion takes place partly in the nozzles and partly in the revolving vanes. A turbine might be so arranged that the expansion would take place entirely in the moving vanes, the guide passages acting merely to change the direction of the steam, but as yet no commercial turbine has been built on these lines.

There are several distinct subdivisions of the two main types of turbine. The simplest form is undoubtedly of the DeLaval type, which consists of several diverging nozzles, expanding the steam from boiler pressure to exhaust pressure, and directing the steam jets onto the vanes of a single wheel. We have seen that the enormous velocity of 4,000 feet per second will be attained in expanding from 150 pounds boiler pressure to $28\frac{1}{2}$ inches vacuum. The speed of revolution must be very high and, although the velocity is greatly lowered as the steam passes through the wheel, it will leave the wheel with a considerable residual velocity which represents, of course, so much lost energy. Fig. 18 illustrates the typical features of this style of machine, the curves showing the relation of its steam velocities and pressures. It will be noticed that the steam pressure is a maximum, and equal to boiler pressure, at the inlet to the nozzle, and will reach the condenser or exhaust pressure at the nozzle outlet, as it impinges upon the vanes of the wheel. Clearance, in this type of turbine, is of small consequence, for the wheel revolves in steam of a uniform pressure, and there can, therefore, be no leakage of steam without work being done. As there is but one wheel revolving in the bath of steam, the friction would

not be very great, were it not that the friction increases very rapidly with the speed, and in this single-wheel type, the speed of the steam is very high. The chief loss will be due to the relatively high velocity of the exhaust steam, and to the friction of the bearings on account of the high rotative speed. To reduce these speeds of rotation to man-

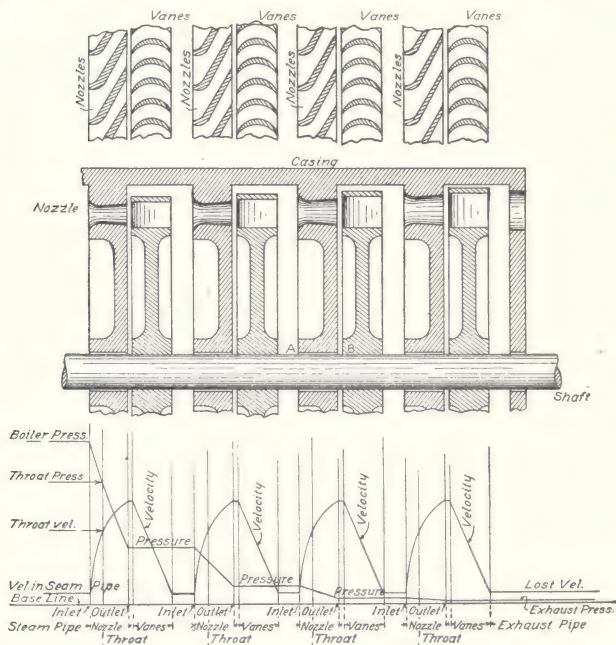


Fig. 19. Features of Multi-Stage Impulse Type, Showing Relations of Steam Pressures and Velocities.

ageable rates, gearing must be used, causing a further frictional loss, or the diameter of the wheel must be abnormally great.

The velocity of the steam at the entrance to the nozzle is that due merely to its flow through the pipe. At the throat of the nozzle, the velocity, as we have previously seen, will be something under 1,500 feet per second, and at the mouth of the nozzle, if it is properly

designed, the velocity will approximate 4,000 feet per second, assuming a boiler pressure of 165 pounds absolute and $28\frac{1}{2}$ inches of vacuum. This high velocity will not be maintained, however, as the steam passes through the revolving vanes, but, at the condenser, will have dropped to a value depending upon the amount of energy absorbed from the steam during its passage through the vanes of the wheel.

If wheels were arranged in successive chambers, so that the steam could be expanded in several steps instead of in one, we should have the essential elements of the Rateau type of turbine. Fig. 19 shows diagrammatically the essential features of this type of turbine, and the relation of velocities and pressures, as before. Each wheel rotates in an independent chamber separated from the next by a diaphragm provided with suitable expanding passages, so that the steam, in passing from the first chamber to the second, will be under conditions similar to those obtaining when passing from the boiler into the first chamber, and again may attain a maximum velocity.

In a four-stage turbine of this sort, the pressure, as shown in the curve in Fig. 19, should be a maximum (boiler pressure) at the inlet to the first nozzle. At the throat of the nozzle, it should be approximately 58% of the initial pressure. During its passage through the first chamber, the steam pressure would be constant, and it would again drop in a similar manner, in passing through the nozzles between the first and the second chamber, the velocity rising with each drop in pressure. With a four-stage turbine, the drop in pressure would be such that one-fourth of the total available heat units would be available in each chamber. The drop in pressure from chamber to chamber would therefore not be uniform, for a given pressure change represents more heat units in the lower than in the higher ranges of pressure. The velocity at the inlet to the first nozzle would again be merely the velocity of flow through the steam pipe; at the throat of the nozzle, approximately 1,500 feet per second, and at the outlet to the nozzle, where the steam impinges upon the vanes of the first wheel, approximately 2,000 feet per second. This velocity will drop as the steam passes through the wheel, rising again on its passage through the next nozzle, dropping again in the next wheel, and so on, the residual velocity as the steam leaves the last wheel being probably less than in the previous case.

somewhat as the steam passes through the first set of running vanes, remains constant as it passes through the first set of guide vanes, again drops in the next set of running vanes, and again becomes constant in the guide vanes, and so on; the velocity of the steam jet is gradually lessened as it passes through wheel after wheel. The drop in velocity in the steam in its passage through any one set of vanes will, neglecting losses, be approximately equal to the total velocity divided by the number of sets of running wheels. In this type of turbine, since the velocity of the steam is gradually decreased, it is evident that if the same quantity of steam is to flow through successive wheels in the same interval of time, the passages must gradually increase in size. The velocity remaining constant in the guide vanes, they may provide passages of uniform section, as shown in Fig. 20, each set of passages, however, being larger than the preceding set.

The principle just described was the original idea claimed in the early Curtis patent, but was subsequently given up for the improved arrangement shown in Fig. 21. This arrangement differs from the other, in that, instead of fully expanding the steam in one nozzle or set of nozzles from boiler to exhaust pressure, the expansion is divided into two or more stages. This turbine contains chambers, just as the Rateau type does, the difference being that in the Curtis, each chamber contains two sets of running wheels and one set

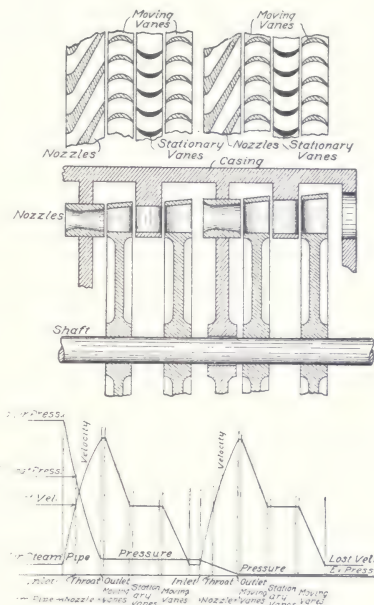


Fig. 21. Features of Turbine Compounding by Pressure Stages and Velocity Steps.

of guide vanes, while the Rateau chamber contains only one wheel and no guide vanes. Turbines of the Curtis type have from two to seven pressure stages, but at the present time, no more than two sets of running vanes are used in each chamber*, although formerly, more sets of running vanes were used. The relation of pressures to velocities shown in Fig. 21 will be evident from the previous ex-

planations.

In the reaction turbines, of which Parsons' is the best known example, the steam, as already stated, gradually expands in passing from boiler to condenser pressure. The velocity rises in the first set of stationary vanes, and drops as the steam does work in the first set of running vanes. The velocity rises again in the next set of stationary vanes, drops in the moving vanes, and so on. Fig. 22 shows the essential features of this turbine and the relation of pressures and volumes.

The stationary guide vanes act just like small nozzles, and allow the

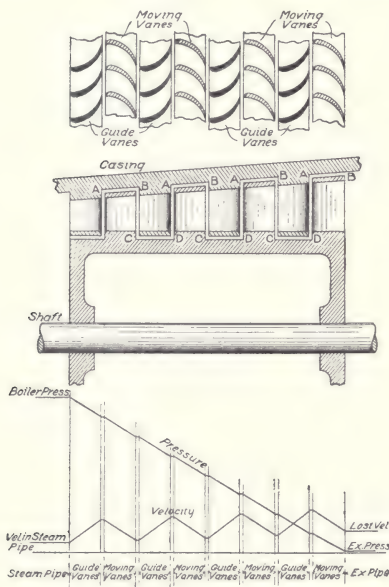
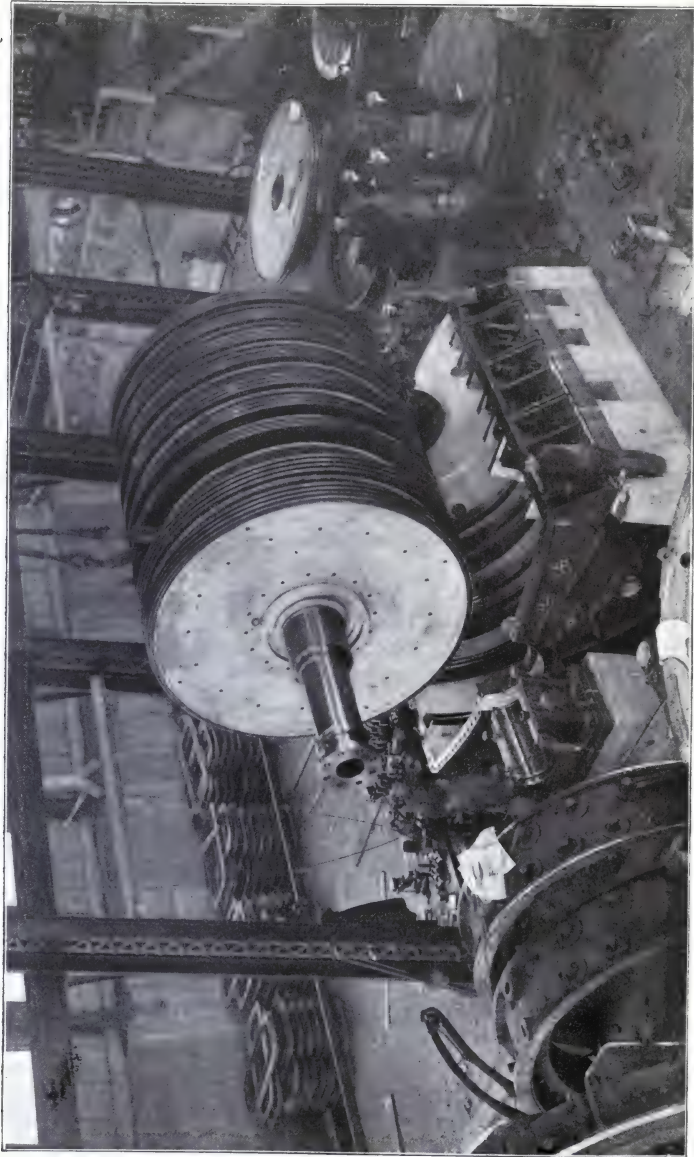


Fig. 22. Features of Reaction Type.

steam to expand and acquire velocity. The moving vanes also allow the steam to expand, and the reaction of this expansion gives an added impulse to the rotating wheel.

It is not intended that the foregoing shall be a description of any turbine, but merely a description of the distinct and elementary features of the action of steam in various types of turbine.

*See foot-note, Page 22.



LOWERING ROTOR INTO CASING
12,000-H. P. Marine Reversible Curtis Turbine
Fore River Ship Building Co.

In the one-stage, compound-velocity turbine, the steam leaves the nozzle at exhaust pressure with a high velocity. If the expansion has been complete, as intended, the pressure remains constant as the steam passes through the turbine, and there is no tendency to leakage. The clearances between the blade tips and the casing can be made as large as convenient, for it requires a difference of pressure to cause steam leakage. If running on vacuum, there would be a tendency for air to leak in around the shaft, and consequently this would need to be well packed.

Here would seem to be a happy solution to the problem of steam leakage, at the same time producing a most compact form of turbine; but, unfortunately, a considerable loss is brought about by steam flowing past the surfaces of both moving and guide vanes and by the large amount of friction, due to the rotation of the many wheels through the steam which fills the turbine. A further serious disadvantage is that an equal amount of work cannot be done in each set of vanes if the entrance and exit angles of the vanes are made equal, as is usually the case. For example, suppose a 4,000 foot steam velocity to be reduced in four wheels, each wheel absorbing 1,000 ft. per sec. Then, if V_1 , V_2 , V_3 , and V_4 represent the respective velocities at the entrance of each wheel, the available energy is

$$\text{for the first wheel, } \frac{W V_1^2}{2g} - \frac{W V_2^2}{2g} = \frac{W}{2g} \times 7,000,000;$$

$$\text{for the second, } \frac{W V_2^2}{2g} - \frac{W V_3^2}{2g} = \frac{W}{2g} \times 5,000,000;$$

$$\text{for the third, } \frac{W}{2g} \times 3,000,000;$$

$$\text{and for the fourth, } \frac{W}{2g} \times 1,000,000.$$

This difficulty will be remedied by increasing the number of pressure stages, and decreasing the number of wheels in each stage to a minimum. With a large number of velocity compound wheels, the work done by the last wheel would be so small that the frictional losses would be too large to make it at all economical. For example, with six wheels, the last wheel would develop only 9% of the power developed in the first. In turbines of this type, by a suitable

design of the nozzles and entrance and exit angles of the vanes, the same amount of steam energy may be abstracted in each pressure stage.

The leakage in this type would be relatively small, only what would pass from stage to stage. This would be comparatively small, because the steam could escape only through the opening where the shaft passes through the diaphragm (*A B* Fig. 19) that separates the two chambers, and with small clearances this could not be large. With a large number of stages, as in the Rateau turbine, leakage in the high pressure end is not all lost, for it has an opportunity to work in the lower stages.

In the reaction turbine, leakage of steam is a most important factor. As the pressure on the two sides of the vane is different, there is a tendency for the steam to escape between the tips of the vanes and the outer casing *A B*, Fig. 22, also between the ends of the guide blades and the rotor *C D*, Fig. 22. As the rotors are of large diameter, a large area is offered for leakage, unless the clearances are kept very small. Here, the steam leaking from the higher pressures, will of course do work on the lower pressure, but at a less efficiency, just as in the Rateau type. The successful turbine of this type requires great nicety of workmanship in order that the clearances may be adjusted to a minimum.

Low-Pressure Turbines. The greatest drawback to improvement in any existing engine plant, or, in fact, in any mechanical installation, has always been the fact that the equipment already installed must be discarded, often thrown into the scrap pile, while still in fairly good condition and capable of doing a considerable amount of work. In the early installation of steam turbines, this was often done, and in order to increase the capacity of the central station, good reciprocating engines were often thrown out and turbines put in their places. It was, however, soon discovered that this, in many cases, was unnecessary, and that the desired increase in power could be had by simply using low-pressure turbines in connection with the existing reciprocating engines. The low-pressure turbine takes the steam exhausted by the engine, slightly above the atmospheric pressure, and expands it to a lower vacuum than could be economically done in the engine.

While the reciprocating engine is highly efficient for utilizing the available energy of steam between boiler and atmospheric pres-

sure, it is relatively inefficient for utilizing the energy of steam in the lower ranges of pressure, especially at pressures below 20 in. vacuum. The steam turbine, on the other hand, utilizes the available energy of steam in the lower more effectively than in the higher ranges of pressure. Since there is about as much available energy in steam below the atmospheric line as there is in steam above it, there is every reason to believe that this combination of engine and turbine will be a most efficient one. In order that the possibilities and limitations may be fully stated, however, it will be necessary to investigate some of the characteristics of steam expansion.

A single cylinder engine with cut-off at, say, one-third stroke, will expand the steam to three times its initial volume, and if it takes steam at 150 pounds gauge pressure, the volume of each pound of that steam before expansion will be approximately 2.75 cubic feet. Now, if this is expanded to three times its initial volume, every pound of steam entering the cylinder will, at exhaust, occupy $3 \times 2.75 = 8.25$ cubic feet. If the expansion has been adiabatic, that is, without the gain or loss of heat, this pound of steam will occupy 8.25 cubic feet of space when the pressure has reached 32 pounds by the gauge, and, under the above conditions, an engine would release at this pressure—a manifest waste.

With one-fifth cut-off and five expansions, the final volume of one pound would be $5 \times 2.75 = 13.75$ cubic feet, and this volume would be reached at about 11.7 pounds gauge pressure. Fig. 23 will illustrate this. The line $b c d e$ is a curve representing the relation of pressures and volumes of steam, as it expands adiabatically from 150 pounds gauge pressure to the atmosphere and beyond the atmosphere into partial vacuum. The total available work in the steam above atmospheric pressure, would be represented by the area of the diagram $a b c h$. The greatest possible work that could be done in the cylinder, cutting off at one-third stroke and exhausting at atmospheric pressure, would be the area $a b c g h$, which shows that a considerable amount of the energy is lost. Even cutting off at one-fifth stroke, the work represented by the area $d e f$ is lost. To carry the expansion of steam in a single-cylinder engine even to one pound above the atmosphere, the boiler pressure must be greatly reduced, or the amount of expansion increased materially. If this engine were made condensing, $m k$ would represent the back-pressure line, and while the

total available energy would be increased by the area $h e m k$, the work in the cylinder at one-fifth cut-off would be increased only by the area $h f j k$, a very small part of the whole. In such case, the gain would probably not pay for the cost of maintaining the vacuum.

In a compound two-cylinder engine taking steam at 150 pounds gauge, the ratio of high- to low-pressure cylinder volumes would be not over 1 to 5, and with cut-off on the high-pressure cylinder at one-third stroke, there would be room for not over 15 expansions; that is, the volume of steam at the end of the low-pressure stroke would be not over 15 times the volume of the steam admitted. Now, if one pound of steam at 150 pounds gauge pressure were expanded to 15 volumes, the result would be $15 \times 2.75 = 41.25$ cu. ft. One pound of steam thus expanded from 150 pounds pressure will occupy 41.25

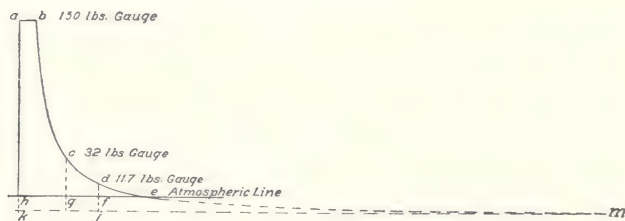


Fig. 23. Relation of Pressure and Volume in Non-Condensing Reciprocating Engine.

cubic feet when the pressure has reached approximately 7.5 pounds absolute, which would correspond to a vacuum of approximately 15 in. In other words, neglecting the condensation and other losses in the cylinder, the ordinary compound engine with Corliss gear (an engine in every way first class), cutting off at one-third stroke, cannot expand steam at 150 pounds boiler pressure lower than to 15 in. vacuum. Any increase of vacuum beyond this point tends only to reduce the back pressure on the piston, and the gain in work is slight, perhaps not enough to pay for the additional work on the air pump, increased size of condenser, and additional circulating water.

Fig. 24 shows, as before, the adiabatic expansion from 150 pounds gauge pressure. If $a b$ represents one volume, $h f$ would represent fifteen. $h f$ will be the back pressure line at 15 in. vacuum, the maximum theoretical work done in the cylinder will be the area

$a b d f h$, and the work lost will be the area $h f m k$. Increasing the vacuum below 15 in. gives only a little gain, represented by the area $h f g k$, although more than in the previous case.

A triple-expansion engine will permit of about twenty expansions; that is, the low-pressure cylinder will contain about twenty times the volume displaced by the piston at cut-off in the high-pressure cylinder. In such an engine, the final volume of one pound of steam expanding from the previous pressure will be 55 cubic feet, and the pressure corresponding to this volume would be 5.5 pounds absolute, equal to about 19 in. vacuum. A condenser giving 24 in. vacuum would allow just about difference enough to give a ready flow of steam

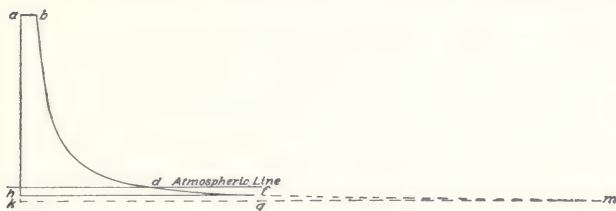


Fig. 24. Relation of Pressure and Volume in Condensing Reciprocating Engine.

from the engine to the condenser. If a greater vacuum is to be used to advantage, the number of expansions must be increased. Even here, increasing the vacuum beyond 19 in. gives relatively little gain in the engine. To expand steam from 150 pounds gauge pressure to 28.5 in. vacuum would require a final volume of 338 cubic feet for each pound of steam admitted to the cylinder, and since one pound at initial pressure occupies 2.75 cubic feet, the steam would have to expand $338 \div 2.75 = 123$ times, approximately. The utter impossibility of such expansion in the triple expansion engine will be evident from the following consideration:

If a triple-expansion engine were to expand the steam to this pressure, with cut-off at one-third stroke, the low-pressure cylinder would have a volume $123 \div 3 = 41$ times that of the high-pressure cylinder, and its diameter would be to the diameter of the high, as 1 is to the square root of 41, or about 6.5. This ratio is not far from three times that found in general practice for such an engine.

and about four times that for a compound engine. Assuming that the low-pressure cylinders are now as large as they can conveniently be made, the complete expansion above outlined would require, in the triple-expansion engine, three low-pressure cylinders of the present size. Radiation loss and friction could easily overcome the theoretical gain; to say nothing of the prohibitive cost and weight of the engine.

Consider the diagram in Fig. 25, which shows, as before, the adiabatic expansion between 150 pounds gauge and 28.5 in. vacuum. The black area represents the available work due to the complete expansion of the steam, in excess of that available in the triple-expansion engine, running under 28.5 in. vacuum. This lost energy is

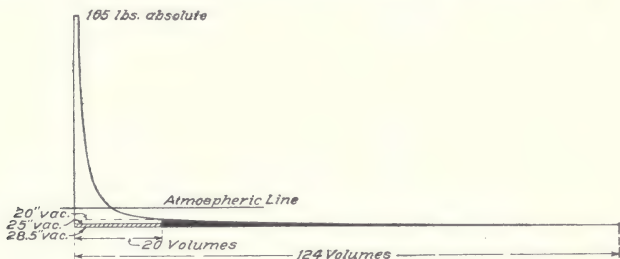


Fig. 25. Energy in Steam not Available for Reciprocating Engine.

about 25% of the total energy available in the steam, or about 35% of the energy available for use in the reciprocating engine with 28.5 in. back pressure. Under the ordinary conditions of 25 in. back pressure, the black area would be augmented by the crosshatched area, making the lost energy about 40% instead of the 35% above. All of this energy is lost by the triple-expansion engine, but can be utilized by the turbine. Low vacuums cause large initial condensation in reciprocating engines, but do not have any disadvantageous effect on the turbine.

The low-pressure turbine can be advantageously used in connection with any reciprocating engine, and their combination will always afford a considerable improvement in economy, and increase the power without increasing the size of the boiler plant. It often happens that engines are operated non-condensing because of

the expense of cooling water, and as we have already shown, the relatively small gain would not pay for additional complications and expense, especially if cooling towers have to be provided. The low-pressure turbine, however, will provide enough additional power to pay for the installation of proper equipment. There are already in existence, plants where low-pressure turbines have been installed in connection with engines previously used as non-condensing, and the output has been practically doubled without increased cost for fuel.

It is readily seen from the previous discussion, that even in a plant in which the engines are operated as condensing engines, a considerable gain can be effected by installing a low-pressure turbine, even though using the same condenser facilities as before. In some ways, it is much easier to maintain a high vacuum in such a combination, because the turbine will take the steam at slightly above the atmospheric pressure, and thus prevent a considerable amount of air leakage, which always takes place through the stuffing-boxes of a low-pressure reciprocating engine.

If saturated steam expands adiabatically from 150 pounds gauge to a pressure of 28.5 in. vacuum, practically half the available energy is developed between the initial pressure and one pound above the atmosphere, and the other half below the latter pressure. It might be said, in explanation, that the work of expansion can be considered as equal to the pressure times the volume; but it is, perhaps, not often realized that the volume of steam will nearly double in expanding from 26 in. vacuum to 28 in., and that, therefore, the available energy is great, although the pressure is low. In most condensing engines, the gain over non-condensing conditions, as determined by actual experiment, does not exceed 30%, even under favorable conditions of steady load. Under average conditions, the gain drops to 25%, and under overload conditions, to a still lower point. In general, a condensing reciprocating engine, if run non-condensing, will carry about 70% of its maximum load, exhausting at, say, two pounds above atmospheric pressure, and, if the steam from such an engine be exhausted into a low-pressure turbine with proper condensing facilities, the latter will develop nearly as much work as was developed by the engine itself, and there will result from the two about 140% of the work which might be expected from the reciprocating engine alone, if run condensing. It is inter-

esting to note that the discussion of Fig. 25 seems to show a possible theoretical gain of about 40% over the engine condensing at 25 in. vacuum, provided the turbine is run at 28.5 in. vacuum.

Fig. 26 shows a study of the possibilities in connection with a Rice-Sargent engine which has been operated for some years in the plant of the General Electric Company at Schenectady, N. Y. This unit operates a 250-v. direct-current generator, and ordinarily runs with a load of 1,200 kw.

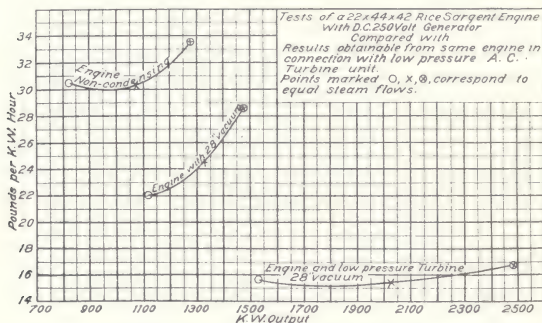


Fig. 26. Curves Showing Economy of Engine with Low-Pressure Turbine.

*The tests were made accurately by weighing condensed steam, the effect of vacuum being determined by holding the steam flow constant, and changing the vacuum. The curves show performance under condensing and non-condensing conditions, and also show what could be accomplished by this engine in combination with a good low-pressure turbine. The rates of gain here shown will seem extraordinary, but they are fairly representative of the possibilities in the average condensing engine plant.

Referring to the curve-sheet, note that the upper curve represents the engine operating non-condensing at 810 kilowatts, the steam consumption being 30.6 pounds per kilowatt. With the load increased to 1,065 kilowatts, the steam consumption is still 30.6 per kilowatt and with the load increased to 1,265 kilowatts, the steam consumption is 33.6 pounds. Operating under these conditions, 1,265 kilowatts is practically the maximum capacity of the unit.

Now, operating condensing with a capacity of 1,140 kilowatts, the steam consumption is 22 pounds per kilowatt; at 1,320 kilowatts, the steam consumption is 24.6 pounds per kilowatt; and operating at 1,470 kilowatts, the steam consumption is 28.8 pounds per kilowatt. Note, however, that the maximum capacity of the unit has been increased from 1,265 to 1,470 kilowatts.

*From a paper by Chas. B. Burleigh, on the "Low-Pressure Steam Turbine."

Now, by the assistance of the low-pressure turbine, vacuum conditions remaining the same, the steam consumption at 1,550 kilowatts is 15.6 pounds per kilowatt; at 2,020 kilowatts, the steam consumption is 15.4 pounds per kilowatt; and at 2,500 kilowatts, the steam consumption is 17 pounds per kilowatt. By this combination, the maximum output of the unit has been increased from 1,265 kilowatts, non-condensing, to 2,500 kilowatts, or from 1,470 kilowatts condensing, to 2,500 kilowatts.

It must not be thought that all this gain can be attained with no compensating loss. In the first place, a surface condenser, to maintain 28.5 in. vacuum, must be about twice the size of one to maintain 26 in., and requires special apparatus that is not only costly, but difficult to maintain. Again, the cost of maintaining a 28.5 in. vacuum is very much more than that of maintaining a 26-in. vacuum, leaving out of consideration the extra cost of condenser and cooling water. After all, it is the dollars and cents that determine the best efficiency, and it is poor economy to obtain the extra power at a greater cost than the returns will warrant. A gain of 35% or more in steam consumption may easily be effected by installing a low-pressure turbine, but the gain in dollars and cents is seldom as great; just what the gain may be, must of course depend upon the local conditions, especially upon the conditions under which the reciprocating engine is operating. In the majority of cases, such installations are worth while, even though used with the usual vacuum.

An interesting application of the low-pressure turbine in connection with rolling mill machinery and other intermittent work, has been worked out by Professor Rateau, and has been made possible by the use of his *steam accumulator*, or *regenerator*. This apparatus regulates the intermittent flow of steam exhausted from the rolling mill engine, let us say, and intended to be used by a low-pressure turbine. The accumulator may consist of a large tank in which are numerous plates over which water can flow, or may contain simply water rapidly circulated by artificial means. As the exhaust steam from the engine enters this accumulator, it spreads out over the exposed water surface, and some of it is condensed if there is an excess of pressure due to more steam being supplied by the exhaust than is being utilized by the turbine. On the other hand, if the turbine utilizes more steam than is supplied by the exhaust, this causes a lowering of the pressure in the accumulator, and a rapid vaporization occurs from the exposed water surfaces, tending to equalize the

pressure. The accumulator thus bears the same relation to the transfer of heat from the reciprocating engine to the turbine that a fly-wheel bears to the transfer of work from the cylinder of the engine to mill shafting. Fig. 27 shows one form of the Rateau accumulator. It must be provided, of course, with a safety-valve, set at a pre-

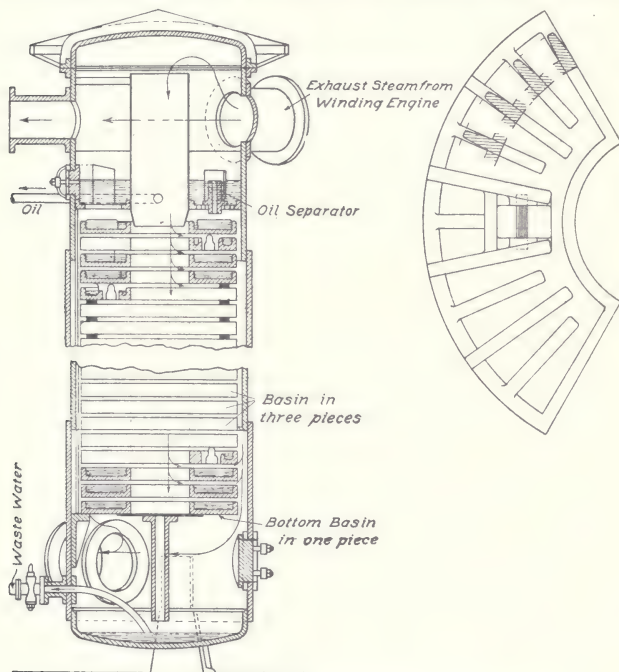


Fig. 27. Interior View of Rateau Accumulator, with Iron Trays.

determined pressure, and is usually provided with a reducing valve from the boiler, so that in case the reciprocating engine should stop for a considerable length of time, steam could still be supplied to the turbine through the reducing valve.

The first apparatus of this kind was installed in 1902, and has been very successful. The first to be installed in the United States was at the Wisconsin Steel Company, in South Chicago. In this plant,

steam first goes to a receiver to take out the shock due to the puffs of the exhaust. From here it passes to the regenerator. The receiver is fitted with baffle plates and drains for water and oil, by means of which they are thus separated from the steam. This accumulator at South Chicago furnishes steam for a low-pressure Rateau turbine which is used to furnish electric power for general purposes.

Installation. The field of the steam turbine is unfortunately limited in its usefulness by two very important factors; first, its relatively high speed of revolution, even when compounded; and, second, its non-reversibility. If, as in marine work, reversing is absolutely necessary, then another turbine, which runs idle ordinarily, with vanes set in the opposite way must be fitted on the shaft. To make this reversing turbine as small as possible, efficiency is sacrificed, but this is of small consequence, for it is used so little. It of course adds materially to the first cost of the turbine and increases the length of the necessary floor space.

The first and greatest field of turbine usefulness is undoubtedly central station work for the generation of electricity by direct-connected apparatus. It also has an important field in driving blowers, centrifugal pumps, etc., where high speed of revolution is essential. In such cases, it has a distinct advantage, for it may be direct-connected, thus doing away with the belting necessary if reciprocating engines were used. The turbine has been suggested to some extent for driving mill shafting, in which case, of course, the speed is belted down from a small pulley on the turbine to a large one on the countershaft, but this appears to offer no particular disadvantage, for in any case belting would be used, as the countershaft would never be run at the same speed as the ordinary reciprocating engine.

In the field of electric generation the turbine to-day has practically superseded the reciprocator. The number of installations is very great, and probably no new central station is now designed for other than steam turbines. In 1906 the Committee of the National Electric Light Association, after an extensive investigation of turbines, reported a wide use of turbines for electric generation, and their figures showed that about 75% of all the turbine units of 500 kw. or over already installed in the United States, were for electric purposes, and that practically only one new central station

abroad had been found installing reciprocating engines. The distinct advantage of turbines for this work is the uniform turning effort, the high speed of rotation permitting the use of a very much smaller generator, and the smaller floor space, requiring less capital outlay in land and engine-house. These features place it in striking contrast with the ponderous slow-moving Corliss engine.

The General Manager of the Metropolitan Street Railway Co. of Kansas City is authority for the statement that in that station six 5,000 kw. units of a well-known make of turbine could be installed in space previously occupied by three 3,000 kw. engine-driven

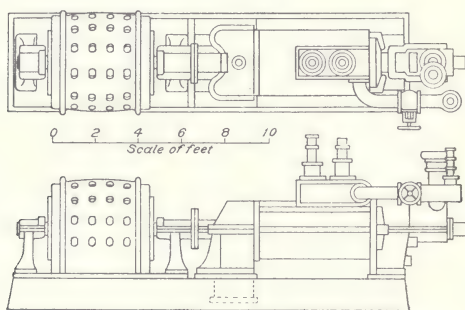


Fig. 28. Plan and Elevation of 500-K. W. Westinghouse Turbo-Generator. This is Same Scale as Fig. 29; Notice Difference in Space Required.

units. Or in other words, 30,000 kw. of turbine power could be put into a building where before only 9,000 kw. of engine power had been possible. This probably is greater than would ordinarily be met with, but the difference in any case is large, the saving in space depending upon the type of turbine. The average horizontal turbine and generator with auxiliary apparatus will occupy about three-fifths of the space needed for a slow-speed, engine-driven generator of the same power, and a vertical turbo-generator somewhat less space than the horizontal.

A further distinct advantage of the turbine is in the fact that, since there are no valves to adjust, the efficiency can be lowered only by wear, and then only slightly; on the other hand, in reciprocating engines, if the valves are not set exactly right, very poor economy

will result, and the opportunities for wear are far greater than in turbine engines. Again, the turbine can use high degrees of superheat because there is no lubricant to burn; there is also little danger of entrained moisture in the steam wrecking the turbine, and the

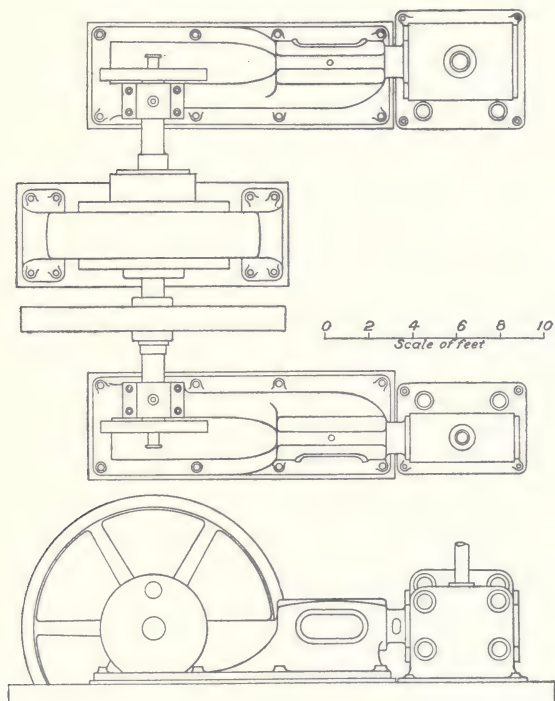


Fig. 29. Plan and Elevation of 500-K. W. Corliss Engine-Driven Generator Set. Compare with Fig. 28.

absence of oil in the condensed steam greatly lessens trouble in the boiler if the condensation is used for feed water. The economy of space was graphically illustrated by Fig. 1, and Figs. 28 and 29 tell the same story but with different types of engine and turbine.

Figures showing the relative space occupied by reciprocators and turbines are of little value unless the size of condenser and condensing auxiliaries are taken into consideration, for, as before mentioned, they may easily be, in the case of the turbine, twice the size of those used with a reciprocating engine of the same power. The apparent saving of space, therefore, may be offset by these auxiliaries. By placing the condensers underneath the turbine, as is frequently done at the present time, not only may a considerable amount of floor space be saved, but the turbine can more readily exhaust into the condenser. As we have already seen, at high vacuum the volume of steam is very large, and the exhaust pipe from the turbine will be proportionally large. It would thus appear that to have the condenser any great distance from the low-pressure end of the turbine would be not only a distinct disadvantage, but offer a considerable practical difficulty.

Turbines, as we have seen, require very much smaller foundations than reciprocating engines of the same power, and these foundations will therefore cost very much less. It is hard to get a direct comparison between turbines and reciprocating engines as a class, because the foundations for high-speed reciprocating engines will not be as massive as for the heavier, low-speed engines. The turbine, occupying less floor space, will require smaller buildings and less land, and this will in a number of cases be a substantial saving in first cost and subsequent interest charges.

So far as the first cost of a generating plant goes, there is at the present time very little difference between those using reciprocating and those using turbine engines. The turbine itself costs more than the reciprocating engine of the same power, but on the other hand the generator for the turbine costs very much less. Again, the condenser and pumps, if high vacuum is to be maintained, will cost two or three times as much as for the reciprocating set, while the cost of erection is decidedly in favor of the turbine. It is not easy to get a direct line on the relative cost of turbine and engine installations, for the figures available appear to vary about as much between reciprocating engines and turbines as might be expected to be found between various installations of reciprocating engines, and undoubtedly turbine installations in some cases cost relatively more than in others. It seems probable that the cost of the turbine is

regulated more by the cost of the reciprocating engine with which it has to compete, than by the actual cost of manufacturing the turbine. All in all, there is likely to be a somewhat less cost of complete installation in favor of the turbine, but the difference will not be large in any case, and in powers under about 100 kw., it is probable that the engine installation is fully as cheap. This does not take into account the value of land and buildings, which in all cases is an important factor in favor of the turbine.

Performance. The losses occurring in the steam turbine consist principally of loss of velocity of the steam itself due to friction in contact with the vanes and guides; friction of the disks revolving through a chamber filled with steam; eddying of the steam jet, due to improper speed of the revolving disks; radiation; bearing friction. The two latter items are not large, and under ordinary conditions would consume less than 2% of the power.

The most important losses are due, first, to the friction of the steam jet against the vanes and guides, which will be approximately proportional to the cube of the velocity of the steam relative to the vanes or guides, and second, to the considerable amount of friction of the disks as they revolve in the chamber filled with steam. This friction generates heat which raises the temperature of the steam and metal parts and thus causes the re-evaporation of some of the condensed moisture. Since this adds some heat to the expanding steam, the expansion is not absolutely adiabatic. The smoother the revolving wheels are made, the less will be this friction, a fact well illustrated by a reported improvement of about 1% in steam consumption which was effected in a well-known make of turbine by making the riveting on the revolving disks perfectly flush. To these losses may properly be added the generator losses which, of course, are a factor of the speed of revolution.

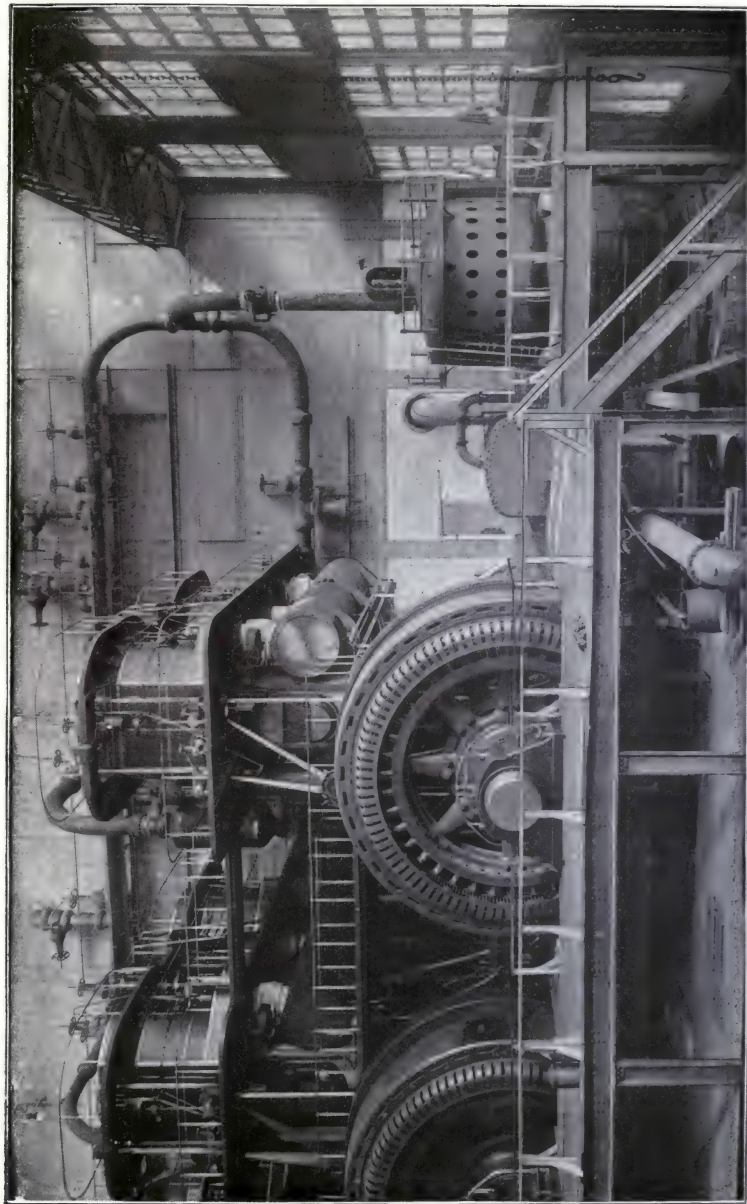
With either reciprocating engines or turbines, the steam economy is much better in large than in small units, and especially is this true of the turbines of the reaction type. In small turbines of this type, the steam friction is high and the leakage large, and this makes it undesirable to build this type of turbine in sizes much below 500 kw. For the impulse type of turbine, these losses are not as important in the smaller powers, and DeLaval, Curtis, and Rateau turbines of comparatively small power can be built to give nearly as good

steam economy as larger turbines of the same type, and can easily excel small reciprocating engines.

The steam consumption of the turbine depends naturally enough upon the vacuum, steam pressure, degree of superheat, variation in load, and variation in speed. It has already been explained that the turbine can utilize the lower ranges of vacuum far better than can the reciprocating engine, but it could not, in all probability, use the higher pressure ranges with as good economy as the best reciprocating engines. If the turbine runs at a vacuum of 27 in., its steam consumption will be practically on a par with that of the reciprocating engine, and it will show a gain of about one-half pound of steam per kw.-hr. for each extra inch of vacuum, below 25 in. But from the saving effected by this one-half pound of steam must be deducted the extra cost of maintaining the high vacuum, if the real economy is desired. Not only can the turbine theoretically utilize the greater vacuum to better advantage, but it has an advantage also in a practical way, because with the reciprocating engine, a very high vacuum cools the cylinder walls and thus causes a relatively large initial condensation, which difficulty is not met with in the turbine, the high vacuum having no detrimental effect. It thus has both a theoretical and practical advantage.

Superheated steam, whether used in the reciprocating engine or in the turbine, will reduce the steam consumption; but in the reciprocating engine, superheating cannot be carried very high, as the cylinder lubricant is likely to be burned, and there will be little condensation in the cylinder to help out the lubrication. The turbine is not handicapped in this way, but nevertheless high degrees of superheat are likely to cause trouble due to unequal expansion in the casing, the temperature at the high-pressure end being so much greater than that at the low-pressure end. This expansion is troublesome, but should be provided for in the design.

Superheat affects the economy of the steam engine in two ways; it carries additional heat units into the cylinder, and lessens condensation. It also helps in the turbine in two ways; it carries additional heat into the turbine, and, being less dense than saturated or moist steam, causes less friction within the turbine, and thus effects a mechanical as well as a theoretical gain. It is generally reported that the gain is 10% for each 100° of superheat, but tests



THE STEAM TURBINE vs. THE STEAM ENGINE

The Curtis steam turbine, with generator, seen in the right background, is rated at 5,000 kw. 6,600 volts, at 500 r. p. m. whereas each of the enormous generators connected to vertical compound engines, shown at the left, has a rated capacity of only 3,500 kw. View taken in Waterside Station of the New York Edison Company, New York City.

which appear to be thoroughly reliable do not seem to bear out this claim. $7\frac{1}{2}\%$ to 8% is a better figure.

The saving in steam will be from 1.5 to 1.75 pounds per kw.-hr. for each 100° of superheat, but the real economy resulting from this superheat will be the difference between the value of this saving in steam and the cost of superheating. The superheating plant costs more, not only for the additional expense of the superheater, but for piping, valves, etc. Cast-steel fittings, and valves with nickel-steel valve stems, are usually required for high degrees of superheat.

The usual steam pressure in turbine work is about 150 pounds gauge. If lower than this, some gain in steam consumption may be had by an increase in boiler pressure, but an increase over 150 pounds does not appear to be productive of great economy. A reference to Fig. 25 will readily show that increasing the pressure above 150 pounds will add very little to the area of available work. Fig. 30 shows the curves of economy of a 30-H. P. turbine at different steam pressures. The gain is less and less the higher the pressure becomes, and is small from 75 to 100 pounds. From 35 to 100 pounds the gain is about $33\frac{1}{3}\%$, but this gain is not due entirely to the rise in steam pressure.

The study of steam nozzles has shown that to use steam efficiently, the nozzle must be properly designed with reference to both the initial and final pressures. Now, if the nozzle on this turbine were designed for 100 pounds pressure, it could neither utilize steam economically at 35 pounds, nor at 150 pounds pressure. To show the real gain due to an increase in steam pressure, it would be necessary to have nozzles in each case that were designed for the specific pressures used. Then, and only then, would the curves show the true gain due to increase in pressure. But a study of Fig. 25 shows that if the theoretical gain is small the practical gain cannot be large. It must, moreover, be borne in mind that a high-pressure plant costs more than a low-pressure plant, and for stationary work very high pressures will not pay. On shipboard, where space and weight are at a premium, it may be good engineering policy to install very high pressures, even though the first cost is greater.

Fig. 31 shows the curves for a 600-kw. Curtis turbine with vary-

ing pressures. In this type of turbine, the same conditions exist as in the previous one, the nozzles being designed for only one pressure.

The economy of a turbine varies with the load, as does the economy of the reciprocating engine, but not perhaps to as marked an extent, and the economy depends of course upon the type of turbine. Turbines like the DeLaval and Curtis admit steam through a number of nozzles which are opened and closed either automatically by the governor or by hand. At normal load, about two-thirds of these nozzles would be open and a 50% overload could then be carried with all nozzles open. In the Parsons turbine, steam is admitted all around the circumference of the drum but the admission is intermittent. For heavy loads the valve remains open for longer in-

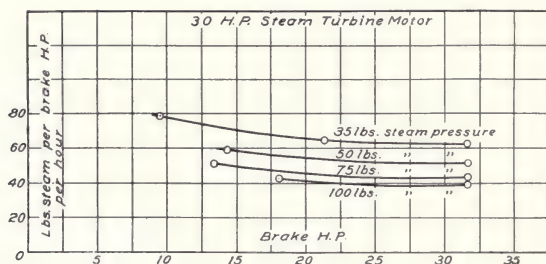


Fig. 30. Curves Showing Economy of 30-H. P. Turbine.

tervals, and when the load is such that the valve remains open all the time, further overloads can be provided for only by resorting to a by-pass which admits high-pressure steam to the second stage of the turbine. In such cases, of course the economy falls off, for the steam does not get the benefit of full expansion. At low loads, there is not a great deal of choice between the different types of turbine, but those that can carry a large overload without opening a by-pass are bound to be the most economical under these conditions.

Overload is taken care of in a reciprocating engine by increasing the cut-off, but, as this reduces the number of expansions, this method is uneconomical. For small ranges of load, the relative economy of turbine and reciprocator are not very different, but the effective range of the turbine is much greater than for the reciprocating engine.

A good turbine will carry 100% overload for a short time and will carry 50% to 60% overload on approximately 10% more steam. Fig. 32 shows characteristic curves of steam consumption at varying loads.

A variation in speed of the turbine within moderate limits does not materially affect the economy. The best speed of the vanes (see Page 17) is half the velocity of whirl ($\frac{1}{2} V \cos \alpha$). Moderate departures from this speed do not materially affect the economy, provided the entrance angles of the vanes are such that the steam jet strikes without shock. The angle of the vane must depend upon the speed, and once fixed, any variation in speed must of course cause the steam jet to spatter and form eddies, a source of material loss. This is entirely apart from the question of whether or not the designed speed of rotation is the most economical. To avoid spattering and eddy losses, the vane angle must change with the speed, which is evidently impossible

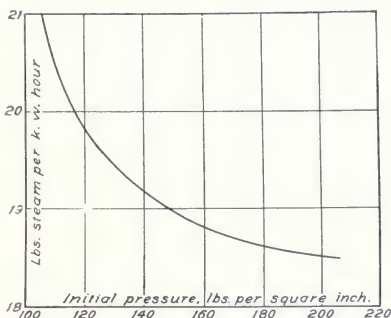


Fig. 31. Economy Curves of 600-K. W. Curtis Turbine with Different Steam Pressures.

A rapid change in load will cause cylinder condensation in a reciprocating engine, so that, on test under steady load, the engine is likely to show up better than it would under service conditions. With the turbine, this is not so. Here there is no such condensation, and the performance under test is far more likely to agree with performance under service conditions. Both types of motor will fall off under service conditions, but if an engine and turbine do equally well under test, under such widely varying conditions as exist in a central station, for instance, the turbine ought to show up better in actual service.

A reciprocating engine is usually designed for a low average load and, therefore, it will permit a relatively large increase in load, but it is generally working on a slight underload, and hence at less

than the maximum efficiency. The turbine, on the other hand, is usually designed for its normal and most economical load, taking care of overload by opening more nozzles at theoretically the same efficiency, or by opening a by-pass at somewhat less efficiency. This should give the turbine a still further advantage at the end of the day's work.

Tests. Tests of reciprocating engines usually give steam in pounds per indicated horse-power per hour, but there being no

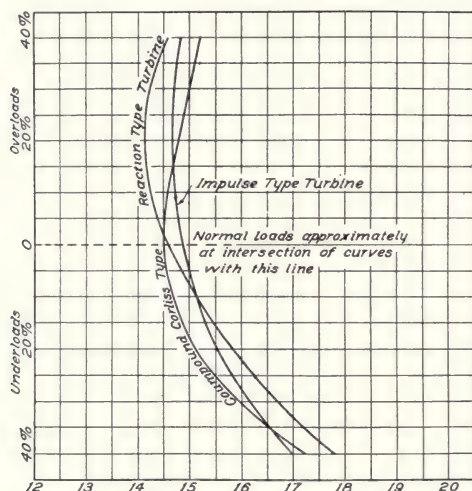


Fig. 32. Economy Curves of Turbines and Compound Corliss Engine with Varying Loads.

indicated horse-power for a turbine, the comparison must be made on some other basis. Brake, or shaft horse-power may readily be obtained for a turbine, and in engine tests, where the brake horse-power has been determined, there is of course opportunity for a direct comparison. However, since engineers are in general more familiar with steam rates per I. H. P., it seems well to consider how a *comparative* I. H. P. may be had for the turbine. Various tests to determine the relation between brake and indicated power

on reciprocating engines seem to show that 92% is a fair figure for a good engine. 92% then of the steam rate per brake horsepower would give the rate per *comparative indicated* horse-power.

The largest field for the steam turbine being central station work, it follows that by far the larger number of turbine tests are quoted in terms of electrical units. It is costly to fit a brake for a large turbine and entirely useless when the power delivered at the switchboard can be read off at once. For electrical work, of course reciprocating engine tests are often quoted in the same electrical units, in which case, there are abundant opportunities for direct comparison.

Suppose, however, that it is desired to compare steam per I. H. P. with a corresponding rate per kilowatt-hour at the switchboard. 1 kw. $\doteq$ 1.34 electric horse-power measured on the switchboard, which is evidently shaft or brake output less losses in the generator. Since the efficiency of a good generator is not far from 95%, the brake horse-power will be equal to the electric horse-power divided by $\frac{95}{100}$. We may say, therefore, in ordinary cases, that

$$\text{B. H. P.} = \frac{1.34 \times \text{kw.}}{.95}$$

since we assume, that $\frac{\text{B. H. P.}}{\text{I. H. P.}} = \frac{92}{100}$,

we have $\text{I. H. P.} = \frac{1.34 \times \text{kw.}}{0.95 \times 0.92} = 1.53 \text{ kw. approximately.}$

Steam per kw.-hr. then, divided by 1.53 would give the steam per *comparative indicated* horse-power per hour, or

$$\text{steam per I. H. P.-hr.} \times 1.53 = \text{steam per kw.-hr.}$$

A turbine using 20 lbs. of steam per kw.-hr. would be about on a par with a reciprocating engine using $\frac{20}{1.53} = 13 \text{ lbs. per I. H. P.}$

In comparing the performance of one engine with the performance of another, or one turbine with another, or an engine with a turbine, pounds of steam per horse-power per hour is generally the rough basis of comparison, but this is very crude and often misleading. For instance, one test may be made with superheated steam and another with saturated or even moist steam, or one may have a higher steam pressure, or the vacuums may be different.

To get an approximately intelligent comparison, all tests should be reduced to a standard degree of superheat, pressure, and vacuum, or better still, if the comparison is between two, correct both to the average conditions of the two. The corrections applied are more or less arbitrary, and it is manifestly unfair to apply them all to either test. If each is corrected for half the difference, a much more reliable comparison is likely to result.

It is generally accepted that the steam consumption will decrease about 8% for each 100° of superheat, about 5% for each inch of vacuum below 28 in., and about 5% for 50 lbs. rise in steam pressure between 100 and 150 lbs., and 3% for similar rise between 150 and 200 lbs. The manufacturer usually gives guarantees of steam rates for various pressures, vacuums, and degrees of superheat. When such figures are available, it is probable that their use would lead to more satisfactory results than if the rough approximations mentioned above were used, but such figures would be correct only for the one individual turbine, and in the large majority of cases the engineer is compelled to use the approximations. They are in most cases fair and satisfactory in the absence of definite data.

To illustrate this method, consider a turbine at 177.5 lbs. (gauge) steam pressure, vacuum 27.3 in., superheat 96° F., consuming 15.15 lbs. steam per kw.-hr., and another using 179 lbs. steam pressure, 29.5 in. vacuum, and 116° F. superheat, consuming 13 lbs. of steam per kw.-hr. The average conditions are 178.2 lbs. steam pressure, 28.40 in. vacuum, and 106° F. of superheat.

The work will appear clearer if arranged in tabular form as in Table I.

TABLE I
Steam Consumption Tests

	TURBINE #1	TURBINE #2	AVERAGE CONDITIONS	TURBINE #1 CORRECTION %	TURBINE #2 CORRECTION %
STEAM BY GAUGE	177.5 lbs.	179 lbs.	178.2 lbs.	0	0
VACUUM	27.3 in.	29.5 in.	28.4 in.	- 5.5%	+ 5.5%
SUPER-HEAT	96°F.	116°F.	106°F.	- 0.8%	+ 0.8%
STEAM / OBSERVED PER KW.-HR.) CORRECT'D	15.15 lbs.	13 lbs.		- 6.3% or	+ 6.3% or
	14.19 lbs.	13.82 lbs.		- 0.96 lb.	+ 0.82 lb.

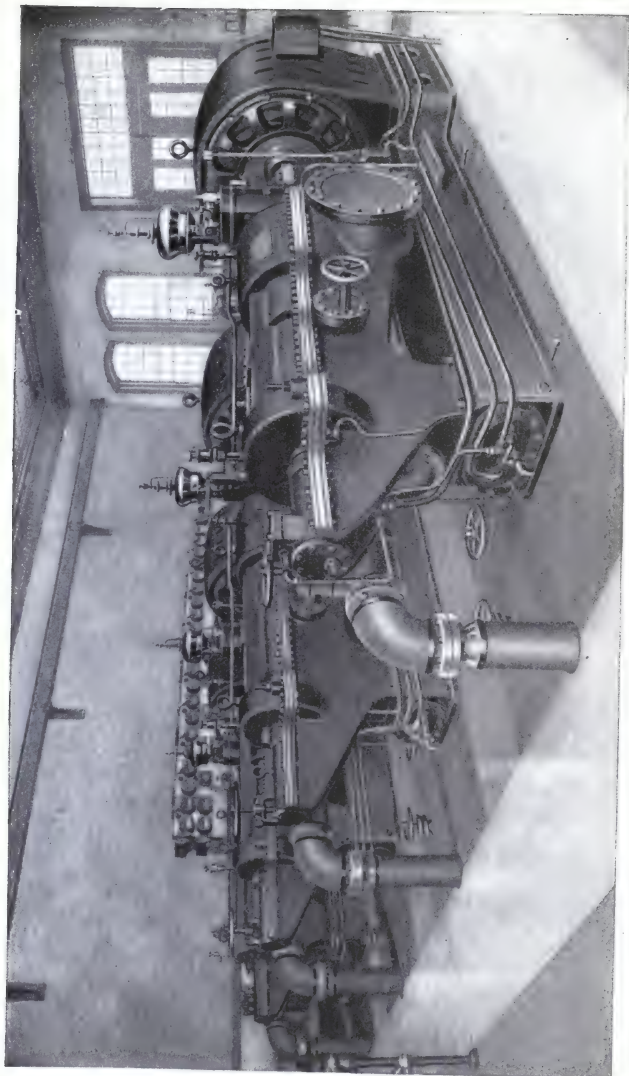
The correction for steam pressure, being only for .7 lbs., is too small to be of consequence in this case. The vacuum correction is ± 1.1 inches, and at 5 % per inch (the decrease in steam consumption for each inch of vacuum, as explained on Page 52), the correction would be $\pm 5\frac{1}{2}\%$. The superheat correction is for 10° , or, as the decrease in steam consumption for 100° of superheat is 8%, this will be $\frac{1}{10}$ of 8% = 0.8%. The sum of these corrections gives $\pm 6.3\%$, making .96 lbs. to be subtracted from turbine #1, and .82 lbs. to be added to turbine #2. The final steam consumptions, then, which should be compared are 14.19 lbs. and 13.82 lbs. instead of 15.15 lbs. and 13 lbs. Turbine #2 appears, therefore, to use about 3% less steam than turbine #1 under similar conditions.

Another and perhaps more satisfactory method of comparison is by means of the heat units used. This computation may be made readily from the steam tables. Using the same tests as given above, turbine #1 uses steam at 177.5 lbs. gauge pressure = 192.2 absolute, at which pressure each pound of dry saturated steam contains 1197 B. T. U. If we allow $\frac{1}{2}$ B. T. U. for each degree of superheat, then, for 96° F. we should add $96 \times .5 = 48$ B. T. U., and each pound would then contain 1245 B. T. U. at admission. If this steam is condensed at a pressure of 27.3 in. vacuum = 1.33 lbs. absolute, each pound of the condensation will contain 80 B. T. U. which will be returned to the boiler in the feed water. The net amount then consumed by the turbine and carried away by the cooling water of the condenser is $1245 - 80 = 1165$ B. T. U. per pound. 15.15 lbs. would represent $15.15 \times 1165 = 17,650$ B. T. U. per hr. or 294 B. T. U. per kilowatt per minute.

Turbine #2 uses 13 lbs. of steam at 179 lbs. gauge pressure and 116° F. superheat, condensing at 29.5 in. vacuum. In this case, each pound of dry steam at admission would contain 1197.3 B. T. U. and 116° F. superheat would add about 58 B. T. U. more, making 1255.3 B. T. U. per pound. Condensing at 29.5 in. vac. = .25 lbs. absolute, each pound of condensed water would contain 27 B. T. U. to return to the boiler in feed water, leaving $1255.3 - 27 = 1228.3$ B. T. U. to be used by the turbine. 13 lbs. would represent $\frac{13 \times 1228.3}{60} = 266$ B. T. U. per min. to compare with 294 in the previous case.

Here, again, the direct comparison is likely to be misleading, unless due account is taken of the difference in conditions. The gain is apparently about 10% in favor of turbine #2 on the heat unit basis taken under the actual working conditions of each, but the fact must not be lost sight of, that turbine #2 is working under more favorable conditions of vacuum and ought to show a much better efficiency. It appears from this discussion that both turbines work under the conditions of design with but little difference in actual economy.

Turbine manufacturers are in the habit of reporting tests of the turbine only, no account being made of the auxiliary apparatus. This is manifestly misleading, for with a 29-in. vacuum, the power consumed by auxiliaries may easily be twice what it would be for a 27-in. vacuum. This extra power and the cost of maintaining it in a measure goes to offset the gain due to the higher vacuum.



THE FIRST TURBINE POWER PLANT IN AMERICA

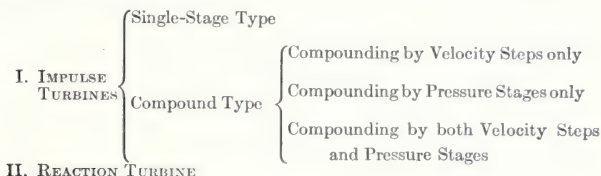
An Installation of Four 400-Kilowatt Turbine Units at Wilmerding, Penna. This Plant Has Been in Continuous Operation Since August, 1899.
Westinghouse Electric & Manufacturing Co., Pittsburgh, Penna.

STEAM TURBINES

PART II

COMMERCIAL TURBINES

*In this description of commercial turbines it will be convenient to classify them as follows:



IMPULSE TURBINES

SINGLE-STAGE IMPULSE TURBINES

Probably the simplest type of turbine is the one with a single stage, that is, a single set of nozzles and a single rotating wheel, but, as already pointed out, the velocity of rotation in a turbine of this sort is so great under ordinary conditions that some device must be employed to reduce the rotational velocity to more convenient speeds. This may be done in two ways.

As has been pointed out before, the feature of importance is not the rotative speed, but the peripheral velocity of the wheel, this being somewhat less than one-half the steam velocity. Maintaining this peripheral velocity constant, turbine rotors of comparatively small diameter may be used, the high rotative velocity being reduced *by means of gearing*; or, *the diameter of the turbine rotor may be increased* the rotative speed thereby being reduced in the same ratio that the diameter of the wheel is increased.

*Many writers group by themselves all turbines using buckets of the Pelton type, but this does not seem to be a proper classification, as it is the action of steam in the turbine that makes it belong to a certain type, and not the style of bucket that is used. Turbines using Pelton buckets may belong to any of the impulse groups.

Both of these methods have been employed in turbines that have been put upon the market, the first method being characteristic of the DeLaval turbine, and the second being employed in the earlier forms of the Riedler-Stumpf machine. The manufacturers of the latter discarded this scheme in their later designs in favor of some sort of compound turbine.

DeLaval Turbines. The turbine designed and developed by Dr. Gustav DeLaval, of Sweden, was among the first to be commercially successful. His first turbine, which was used to run the famous

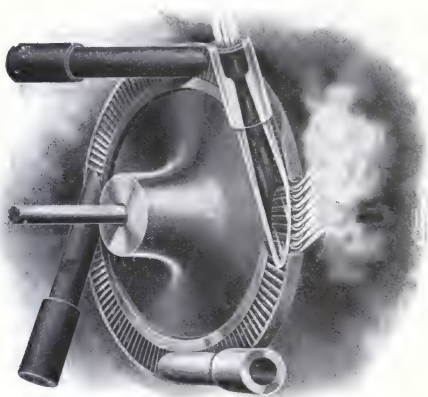


Fig. 33. Principle of Operation of DeLaval Steam Turbine.

DeLaval cream separators, was of the pure reaction type, similar in action to the old Hero engine. This turbine was not economical in steam consumption, but, as it was used for very small powers only, this factor was not important and, commercially, the machine was very successful. This success led to the desire to build larger turbines, and in their development the reaction principle was abandoned.

The essentials of the motor element of the DeLaval turbine are illustrated by their familiar trade mark, shown in Fig. 33. They consist of a rotating disk, having vanes on its periphery; a number of nozzles in which the steam is expanded from boiler pressure to the

pressure in the exhaust chamber and delivered in a jet against the vanes; a long slender shaft to which the rotating wheel is fixed, so arranged that at high speeds the rotating element can revolve about its own center of gravity* instead of its geometrical center; and a set of reducing gears to reduce the high rotative speeds to the desired amount. It is an impulse turbine with a single wheel, carrying one row of buckets, and is a single-stage turbine in all respects. The steam is directed against the vanes from nozzles with flaring sides, so designed as to give it the maximum velocity and to expand it within the confines of the nozzle to the pressure in the exhaust chamber, thus transforming all of the heat energy of the steam into kinetic energy. The nozzles deliver the steam jets at the side of the wheel, and for a maximum efficiency should make as small an angle as possible with the plane of rotation.† In the DeLaval machine this angle is 20° .

For small turbines, the entrance and exit angles of the vanes are 32° , increasing to 36° for the larger sizes. Under these conditions, the best peripheral velocity will be about 1900 feet per second when the velocity of the steam issuing from the nozzles is 4000 feet per second. In most impulse turbines the peripheral velocity varies from 1400 in the larger sizes to 500 in the smaller sizes. These speeds are high, even for turbine work, and necessitate the solution of very interesting engineering problems. These velocities, with the diameters used for DeLaval machines, mean revolutions of about 10,600 per minute in the larger sizes, and 30,000 per minute in the smaller sizes, these speeds being reduced by helical gears to approximately 900 and 3000 revolutions per minute respectively.

This gearing, in the small-sized turbines, consists of a pinion and a single gear, but in the larger-sized turbines there is a single pinion with a gear on each side. This method has the advantage of distributing the load half on each gear, thus lowering the pressure on the teeth and eliminating side pressure on the bearings of the flexible shaft. One disadvantage in the large size, however, is that there are two working shafts, each connected to a working unit. There is also not only the large amount of friction in the gears but also the

*Lack of uniformity in the density of the steel might cause the center of gravity of the wheel to be outside of its geometrical axis.

†See Page 17, Part I.

friction due to the double set of bearings throughout, and the losses that attend the performance of work in two small units rather than in one larger unit.

Nozzles. The wheel is enclosed in a casing in which the nozzles are set, these nozzles being opened and closed by means of hand valves. A detail of the nozzle and valve is shown in Fig. 34. *A* is an annular space in the casing acting as a steam chest, *C* is the valve which permits opening or closing of the nozzle, and *B* is the nozzle itself. The nozzle is fitted into a taper hole in the casting and drawn into place by a nut.

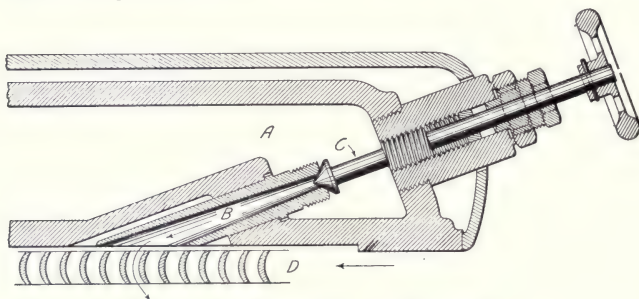


Fig. 34. Detail of Nozzle of DeLaval Turbine.

The design of the nozzle depends naturally upon the pressure used, the degree of superheat, and the vacuum or back pressure. The nozzles being easily removed, it is apparent that a turbine can readily be altered to meet different conditions by inserting new nozzles. A condensing turbine is often equipped with an extra set of nozzles designed for non-condensing conditions, which may be used with better economy in case the vacuum fails.

There are usually two to twenty-four nozzles in the casing, and the power developed at any time is naturally proportional to the number of nozzles in operation. The clearance between the wheel and the nozzle is about one-eighth of an inch. The clearance between the tips of the blades and the casing is not a matter of importance, for there is no tendency for steam leakage, the pressure in all parts of the casing being practically the same as the back pressure. This clearance, therefore, may be whatever practical conditions require.

Fig. 35 is the exterior view of the turbine and generator, showing nozzles and valves set in the casing. By inserting nozzles in the holes which are shown plugged in the figure, a greater power could be obtained.

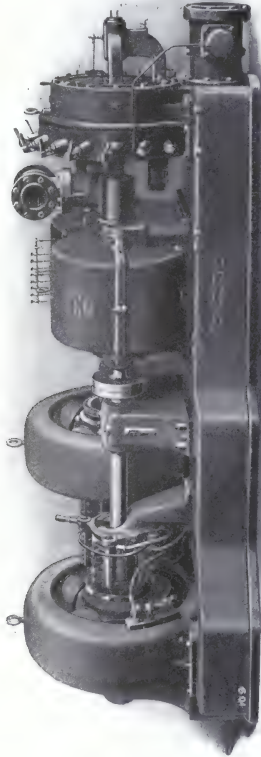


Fig 35. DeLaval Turbo-Generator, 300 H. P., 200 K. W.

Vanes. The vanes are of the crescent shape common in impulse turbines. They are made of drop-forged steel which resists erosion, and have bulb shanks, as shown in Fig. 36, which are driven into

place. The outer ends of the vanes fit closely together, thus forming a continuous ring which prevents any movement at the ends of the vanes.

Steam at high velocities, especially if wet, is liable to cause appreciable wear on the vanes, the wear being practically all on the entrance side; but it is not very great, and tests of a 100-horse-power turbine have shown that wear on the buckets could be as great as one-sixteenth of an inch without increasing the steam consumption more than 3%, according to the report of the manufacturers.

Wheel. At the very high speeds employed, centrifugal forces are enormous; hence, special high-grade nickel-steel must be used in the manufacture of the rotating elements. This steel is said to be

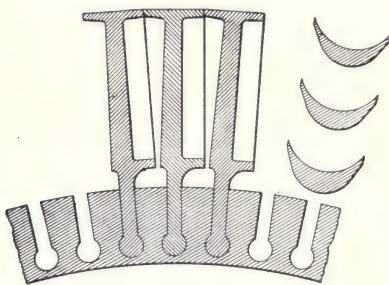


Fig. 36. Method of Attaching Vanes in DeLaval Turbine.

high in carbon and to possess a tensile strength of approximately 135,000 lbs. per sq. in. The wheel is shown in cross-section in Fig. 37 and is designed to be of uniform strength throughout, except that just below the rim a narrow annular groove is turned purposely to make this section weak, for the following reason:

Centrifugal force increases as the square of the speed, and, if the safety devices fail to work, the rotating wheel must ultimately burst. The reduced section near the periphery of this wheel makes the stresses at this point approximately 50% greater than elsewhere, and yet, at normal speeds, this will be perfectly safe, as the factor of safety is between four and five. Now, since the centrifugal force increases directly as the square of the number of revolutions, the stresses at the weakened point, when the speed is double, will be four times as great, that is, about equal to the ultimate strength of the material. The rim will therefore burst and fly into many small pieces, doing but little damage, as the casing is made heavy enough to restrain these fragments. When the rim flies off, the stresses in the main portion of the wheel are thereby greatly reduced, and no further



CURTIS TURBINE DRIVING ALTERNATING-CURRENT GENERATOR.
General Electric Company.

damage can ensue. Wheels without this weak section have burst under test into a few large pieces which have possessed enough energy to break through a 2-in. cast-iron casing.

On each side of the wheel are hubs extending into cylindrical openings in the casing. These are known as *safety bearings* and work with slight clearance under ordinary conditions. Should the rim burst, the wheel would at once become unbalanced, and the resulting eccentricity of the center of gravity would cause the wheel and

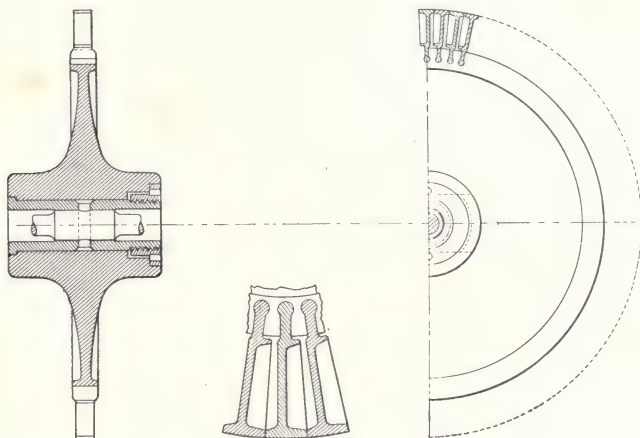


Fig. 37. Method of Mounting Wheel of Small DeLaval Turbine.

shaft to rotate off-center, bringing a considerable pressure of the hub against these safety bearings. These, acting as a brake, together with the absence of further impelling forces due to the loss of rim and buckets, will quickly bring the rotating wheel to a stop.

For small wheels, a bushing is fitted and shrunk to a short swelling on the shaft, and in addition is pinned in place. The hub of the wheel is bored to fit this bushing and it, together with the shaft, is drawn into place by a nut, as shown in Fig. 37. The wheel may readily be removed from the shaft by loosening the nut.

For large wheels, such a construction is not desirable, because a wheel with a hole in the center is not nearly as strong as one without such a hole, and, in the larger sizes of turbine, the strength of the wheel is an exceedingly important factor. The hub, therefore, in such a wheel is solid, but is recessed to fit the flanged end of the shaft, as shown in Fig. 38. The recess is tapered one-half inch to the foot to fit the shaft, which is securely bolted in place as shown.

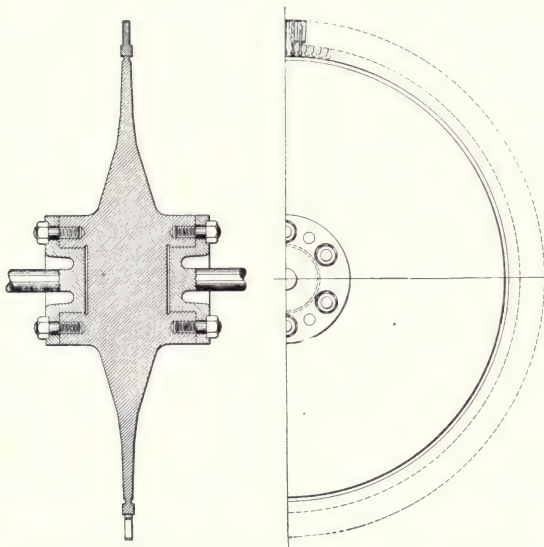
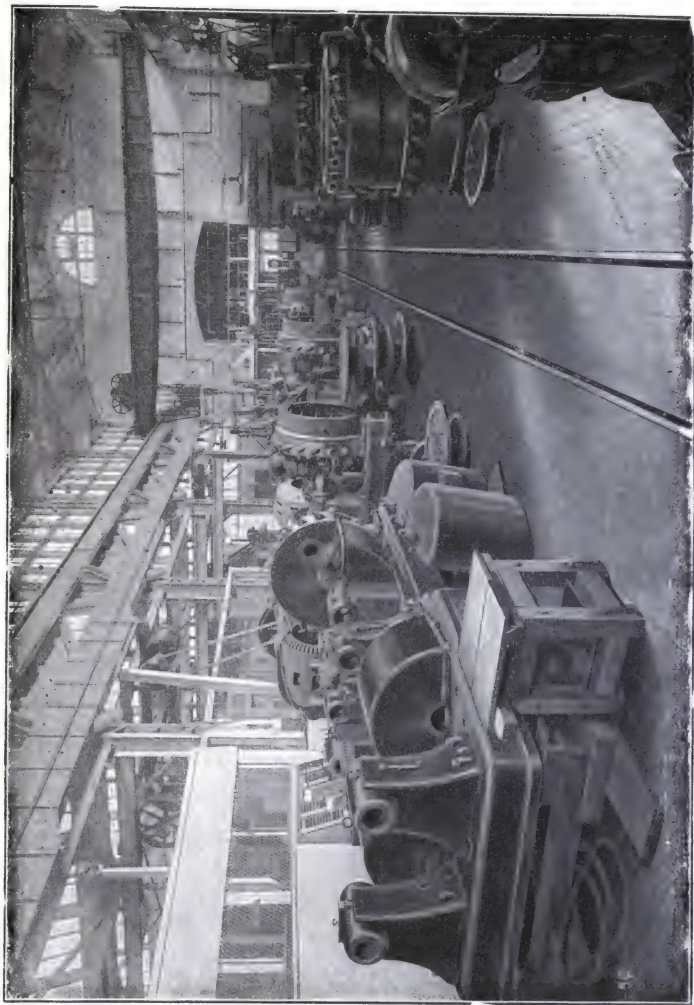


Fig. 38. Method of Mounting Wheel of Large DeLaval Turbine.

The pitch circle of the vanes is about 4 in. in diameter for a 7-H. P. turbine making 30,000 R. P. M.; about $8\frac{3}{4}$ in. for a 30-H. P. turbine making 20,000 R. P. M.; about $19\frac{3}{4}$ in. for a 100-H. P. turbine, making 13,000 R. P. M.; and 30 in. for a 300-H. P. turbine making about 10,600 R. P. M. The rim of the wheel is drilled parallel to the shaft, with cylindrical holes milled out, as shown in Fig. 36, to hold the bulb shanks. This makes a strong construction, and the vanes are easily replaced if necessary.



ERECTING SHOP FOR LARGE STEAM TURBINES.
De Laval Steam Turbine Co.

Shaft. When a body is rotating at high speed, it must be very carefully balanced, by distributing the material symmetrically about the center of rotation. If the center of gravity of the rotating mass is not absolutely at the center of the shaft, a vibration more or less

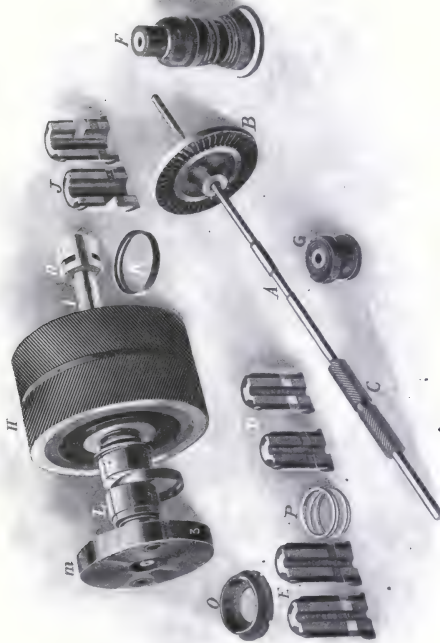


Fig. 39. Working Parts of DeLaval Turbine.

serious will be set up, because a rotating body tends to rotate about its own center of gravity instead of its geometrical center, thus causing a pressure alternately on one side or the other of the bearing. For speeds of 3000 R. P. M., which is common in compound turbines,

the wheels can be balanced on knife edges, the wheel disks being drilled at certain points until they become perfectly balanced. It is reported that careful work in this matter will ensure the center of gravity of the wheel being within $\frac{.004}{1000}$ of an inch of the geometrical center. Small as this error may be, it would be prohibitive at the high rotative speeds used in the DeLaval turbine; hence the adoption of the long, slender shaft on which the wheel is mounted. This bends slightly, and allows the wheel to rotate about its own center of gravity without vibration. This feature is distinctive of the DeLaval machine. The relatively small diameter of shaft is astonishing, being but a little over $1\frac{1}{4}$ in. at its smallest section for the 300-H. P. turbine.

Gears. The speed-reducing gears are in the ratio of about ten to one; *i. e.*, if the turbine rotor has a speed of 30,000 R. P. M., the larger gears have a speed of 3000 R. P. M. At the desired place, a swelling on the shaft is provided, in which the pinion teeth are cut. In the smaller sizes only one large gear is used, but in the larger machines there are two large gears, one on each side of the pinion. The teeth are cut spirally at an angle of about 45° , as shown in Fig. 35, and have double sets of teeth at 90° to each other.

These reduction gears are fine examples of engineering and mechanical skill, as only the best of work would stand up under the high speeds of rotation. The shaft on which the pinion is cut is of nickel-steel, but the gears are made of soft steel low in carbon. They have a peripheral velocity of about 100 feet a second, and if kept free from grit, will run for a long time with little or no wear. These gears were originally made of bronze, but this has proved unsatisfactory, as crystallization of the bronze developed, which resulted in the fracture of the teeth after a few years' continuous use.

Fig. 39 illustrates the various working parts of the DeLaval steam turbine. *B* is the rotating bladed wheel, *A* the long flexible shaft, *C* the pinion cut on the shaft, *H* one set of reducing gears, and *M* the flange for connection to the working unit. Fig. 40 shows a sectional view of a complete turbine and connections for a single working shaft.

Bearing. On the governor end of the shaft is a spherically-seated, self-aligning bearing, provided to give greater flexibility to the shaft, and to take up the thrust due to the action of the steam on the blades. The latter is a relatively unimportant matter, for the

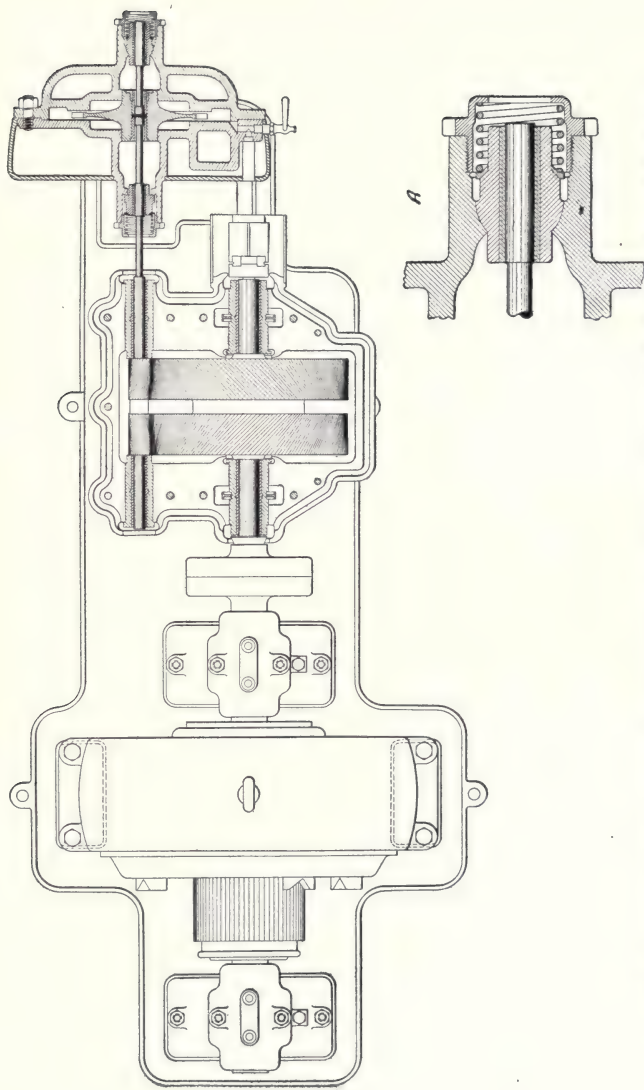


Fig. 40. Sectional View of Complete DeLaval Turbo-Generator. A is the Spherically Seated Bearing with Spring and Cap.

steam is completely expanded in the nozzle, and the pressure is, therefore, practically the same on both sides of the vanes. This bearing is held in place by a spring and cap, shown at A, Fig. 40.

In the high-speed bearings, ring oiling is not satisfactory, and the oil is supplied to shallow spiral grooves. The oil filters through wicking in a sight-feed lubricator and drips upon these bearings. On the gear shafts, the bearings are provided with the ordinary ring oiling devices. All bearings are carefully lined with white metal.

Riedler-Stumpf Turbine. The first turbine developed by Professors Riedler and Stumpf was of the single-stage type, both pressure

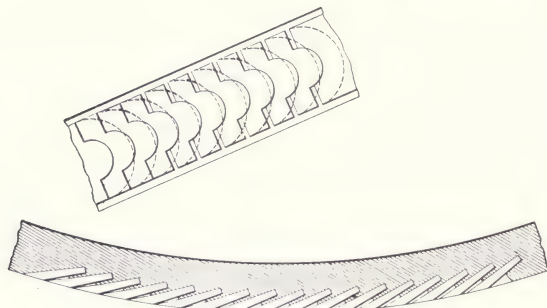


Fig. 41. Milled Buckets in Riedler-Stumpf Turbine.

and velocity, like the DeLaval, but with this radical difference—a wheel about ten times as large in diameter as the DeLaval wheel was used and, therefore, the same peripheral speed was obtained with about one-tenth as many revolutions. The reduction which DeLaval accomplished by means of gears, Riedler and Stumpf accomplished by increasing the diameter of the wheel. By this reduction in the number of revolutions, the error in balance, which, it is claimed, could be brought to less than $\frac{1}{100}$ millimeter, was rendered insignificant.

Their wheels were said to be made of 10% nickel-steel with 135,000 lbs. tensile strength, and were $6\frac{1}{2}$ to 9 ft. in diameter, revolving about 3000 R. P. M. for machines of 2000 to 3000 H. P. Their single-stage turbine did not meet with general favor, and was usually compounded either by pressure stages or velocity steps, but a description of it will nevertheless be valuable.

Instead of using vanes of the DeLaval type, U-shaped buckets were milled in the face of the solid wheel, overlapping one another as shown in Fig. 41. The steam jet impinged upon the buckets—not on the side of the wheel, as in the DeLaval type, but directly upon

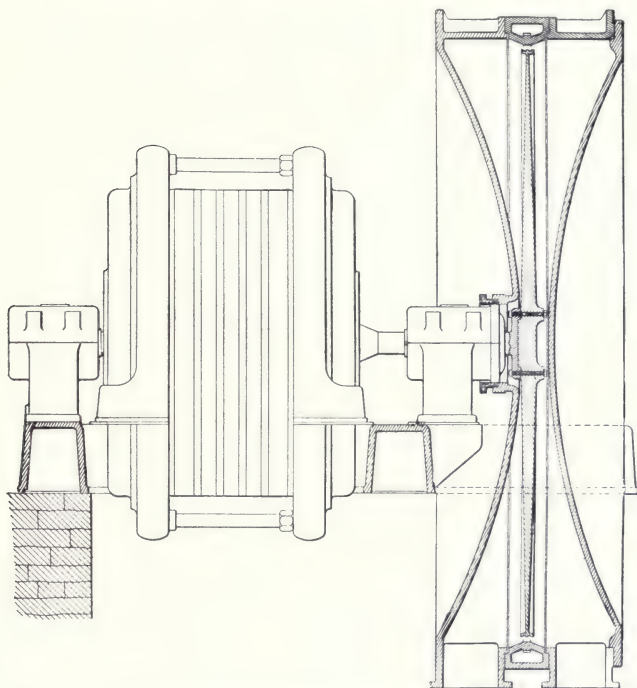


Fig. 42. 20-H. P. Riedler-Stumpf Turbine and Direct-Connected Generator.

the face of the wheel—thus permitting a more nearly complete reversal of the steam jet and, other things being equal, a higher efficiency. It will be recollected that if the jet is delivered to the vanes at the side, and at entrance and exit makes an angle with the plane of rotation, the velocity of the jet* is divided into two components. The velocity

*Page 17, Part I.

of whirl, which is equal to $V \cos \alpha$, α , usually being 20° to 35° , is the only component that produces a rotative effort.

The nozzles were made with a square instead of an elliptical section at the outlet, and were arranged at regular intervals about the casing, as in the DeLaval turbine. With a given size of wheel, the power was increased by increasing the number of these nozzles until steam injection took effect upon the entire periphery of the wheel.

There being but one rotating wheel, it overhung the shaft bearing, thus passing through the casing on one side only, requiring but one stuffing box, and, therefore, giving a comparatively small bearing loss. A 20-H. P. turbine of this type with a direct-connected dynamo, is shown in Fig. 42. Fig. 43 shows details of the wheel. This wheel is fitted with double buckets, which were generally used on the large sizes. A 5000-K. W. turbine of this type would require a wheel 20 feet in diameter, admitting steam to the whole periphery, and making 1500 revolutions per minute. More details of the Riedler-Stumpf turbines will be described in connection with the compound turbine.

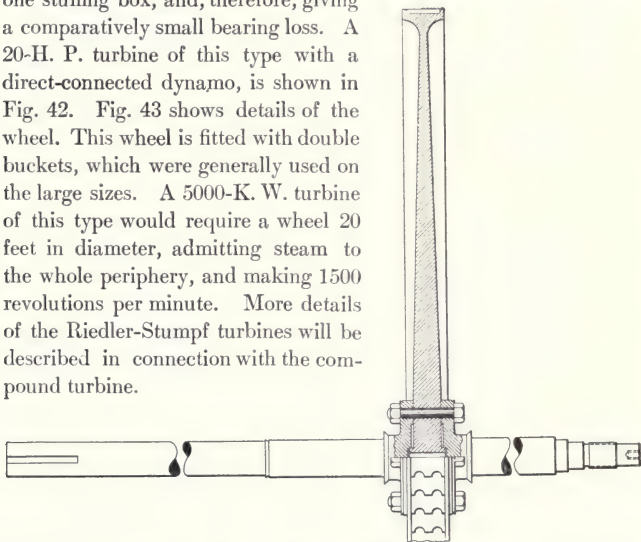


Fig. 43. Detail of Wheel of Riedler-Stumpf Turbine.

COMPOUND IMPULSE TURBINES WITH VELOCITY STEPS

It has been shown that steam may be fully expanded to the back pressure in a single nozzle, and the kinetic energy absorbed by passing the jet through several sets of revolving wheels or vanes in succession, each taking out part of the velocity. To employ velocity steps, some sort of reversing device must be arranged to bring the steam back,

either onto another bucket of the same wheel or onto a bucket of an adjoining wheel attached to the same shaft. The former method was adopted in the Riedler-Stumpf turbine, the latter in the Curtis. In either case, a simple and compact turbine is the result; but the

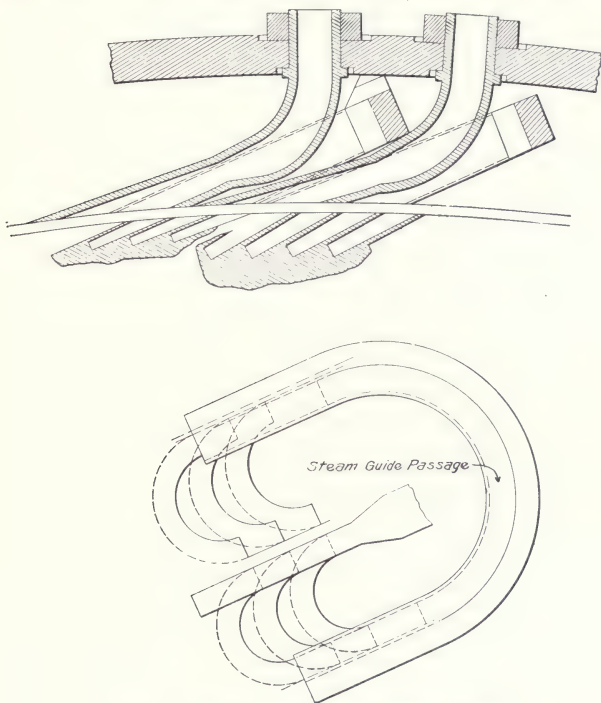


Fig. 44. Double Buckets in Riedler-Stumpf Turbine.

type has disadvantages already enumerated in Part I, which limit the number of velocity steps that can be economically used, to three or four at the outside. Since the work derived from the fifth action of the steam would theoretically be only $\frac{1}{9}$ of that derived from the first action,* and might easily be

*Page 31, Part I.

consumed in additional friction, it is customary to allow the steam to act no more than four times. Single-stage turbines are not considered practicable in sizes above 200 H. P. or 300 H. P., it

being more economical in such cases to reduce the steam velocity by using another pressure stage.

Curtis Turbine. The earlier forms of Curtis turbine were of the single-pressure stage type with several velocity steps, and the smaller turbines now made by the General Electric Co. are after this pattern. 35-K. W. sizes and smaller have a single-pressure stage with three velocity steps, that is, three sets of rotating vanes with two intermediate sets of guide vanes. The details of construction are in all ways similar to those of the ordinary form of Curtis turbine, which is compounded both by pressure and velocity and will be described under that heading.

Riedler=Stumpf. Large powers of the simple impulse type required either abnormally large wheels or too high speeds of rotation and it was, therefore, frequently more convenient to extract the velocity from the steam



Fig. 45. Double Guide Vanes.

jet in two steps. For powers larger than could be dealt with in the single-stage type, the steam passed successively through buckets of the same wheel, and for still larger powers, pressure stages were employed as well as the velocity steps. The compound velocity turbines developed by Professors Riedler and Stumpf had wheels and buckets of the general type described in connection with their simple impulse turbine. The device employed to reverse the direction of

the steam and bring it back again to other buckets on the same wheel was clearly described on Page 9, Part I, to which the student is referred.

In one type of their turbine, the buckets were double, a small bucket on one side of the wheel being for initial admission, and, since part of the steam velocity was abstracted, it was necessary that, as the steam returned, it should enter a larger bucket which was provided on the other side of the wheel, as shown in Fig. 44 and Fig. 45.

Another device of Riedler and Stumpf for reducing speeds of rotation was the employment of counter-running wheels. The

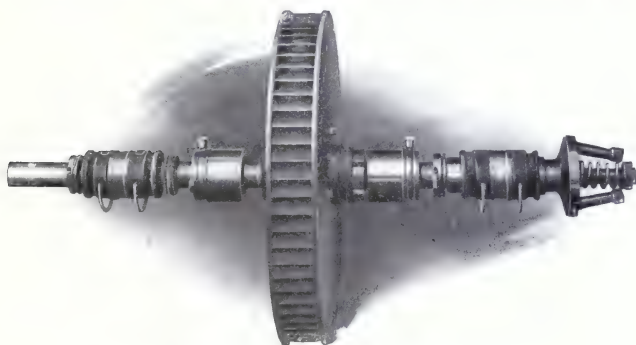


Fig. 46. Rotor and Shaft of Terry Single-Stage Turbine.
Terry Steam Turbine Co.

guide vanes were buckets cut on a wheel which, instead of being stationary, was free to revolve in a contrary direction. Thus, the absolute velocity of each wheel would be half the relative velocity of the two wheels. In a turbine of this type, besides the obvious objection of rotation in two directions, the wheel of initial admission would do more work than the counter-running wheel, because the work absorbed would be in proportion to the difference of the squares of the steam velocities at entrance and exit, and the higher velocities would naturally exist in the first wheel.*

Terry Turbine. The turbine developed and now built by the Terry Steam Turbine Co., of Hartford, Conn., is, in sizes up to 200-

*Page 31, Part I.

H. P., of the single-stage, compound-velocity type. The buckets are U-shaped, milled in the face of the wheel, overlapping one another something like the single bucket arrangement of the Riedler-Stumpf

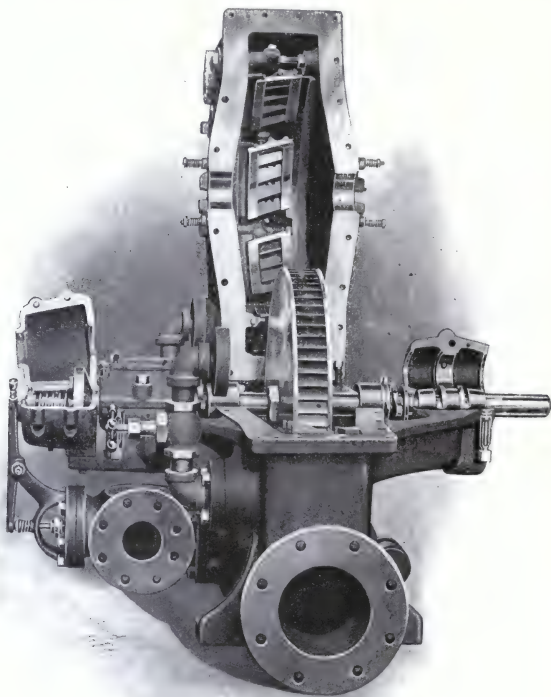


Fig. 47. 110-H. P. Terry Steam Turbine, Opened.

machines. Fig. 46 shows the rotor of a Terry turbine. The steam is expanded in the nozzles to within about one pound of back pressure. As it leaves the nozzles, it impinges upon one side of the bucket, reversing through 180° . As it leaves the first bucket, it enters a

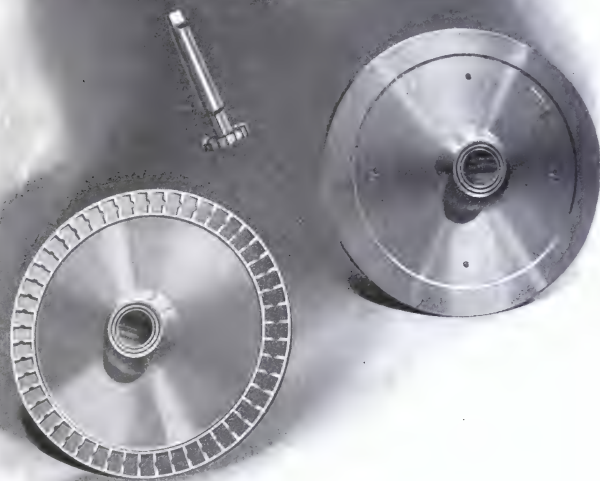


Fig. 48. Wheel of Sturtevant Turbine. The Buckets are Milled from the Side of the Wheel.

similar bucket attached to the casing, which reverses its direction through 180° and causes it again to impinge upon another bucket of

the wheel, and so on until the velocity is all absorbed. The reverse buckets are arranged in groups (usually of four), one group for each nozzle, the steam being returned to the wheel as many times as there are reverse buckets in each group. Fig. 47 clearly shows these buckets on the inside of the lifted casing. A crescent-shaped hole may be seen cut in the bottom of each reverse bucket. These holes release

a part of the expanded steam and thus reduce the volume in proportion to the lessened velocity, as otherwise there ought to be successively larger passage areas.

There are usually four nozzles to a turbine, each being controlled by a hand valve so that the power may readily be regulated. The main bearings are of the ring-oiling type. As the weight of the rotor is comparatively small and the speed of revolution 1250 R. P. M. for 200 to 300 H. P., large sizes offer no practical difficulty.

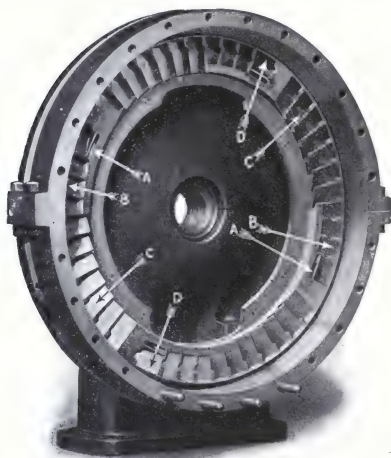
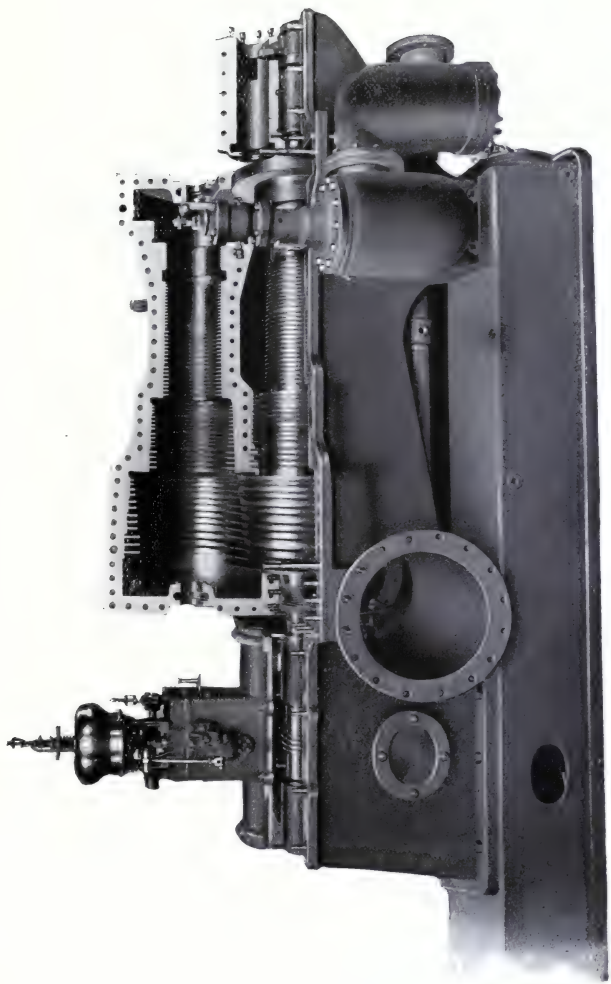


Fig. 49. Sturtevant Turbine Showing Reverse Guides in Casing.

Sturtevant Turbine. The B. F. Sturtevant Company, of Hyde Park, Mass., is building a turbine in small sizes to drive electric generators and blowers. In sizes of 100 H. P., or less, these turbines have a single-pressure stage, using the steam over and over again on the wheel in much the same manner as is done in the Terry turbine. Powers of 200 H. P., or over, would be built with two or more pressure stages. The wheel is a single forging of open-hearth steel. The buckets, which are the U-shape type, are cut from the solid rim by a milling machine.



600 HORSE-POWER · TURBINE · OPEN FOR INSPECTION.

The earlier turbines have buckets cut on the side of the wheel, as shown in Fig. 48. Steam entering the outer edge of these buckets, passes through the buckets into stationary reverse guides in the casing shown in Fig. 49. At *A* are inserted the nozzles, which are of the ordinary expansion type with elliptical openings. The guides are of two types; about four are U-shape like the buckets and return the steam to the wheel again, returning it as many times as there are return buckets; the others, shown at *C*, are cut open at the inner edge in such a manner that the steam, instead of returning to the wheel, is exhausted into the middle of the casing and there allowed to pass out. To avoid a troublesome end thrust in this machine, buckets are cut on both sides of the wheel, thus equalizing the side pressure.

The later machines built by the Sturtevant Co. have the buckets on the outer rim much like those shown in the illustrations of the Terry turbine, but the action of the steam is not different from that in the earlier type. Fig. 50 shows this style of turbine.

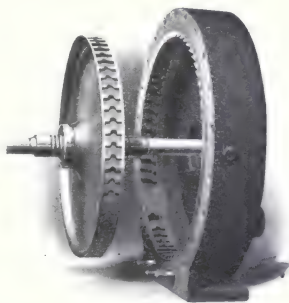


Fig. 50. Sturtevant Turbine with Buckets Cut on Outer Rim.

COMPOUND IMPULSE TURBINES, PRESSURE STAGES

In the discussion of compound turbines in Part I, it was shown that the available head could be divided into several stages, thus making the steam velocity from stage to stage relatively small, and permitting smaller speeds of revolution. Turbines of this type are, in principle, like a number of DeLaval wheels on the same shaft. They consist essentially of a casing which supports a number of diaphragms, dividing the interior into separate cells, in each of which a single impulse wheel containing the vanes is free to revolve. Each stage or element comprises a rotary wheel and a set of nozzles, or distributing vanes, which guide the steam from one chamber to the next and direct it at the proper angle onto the vanes of the wheel in

the following chamber. These passages may or may not be of the diverging type, depending upon the drop of pressure from stage to stage. In all machines of this type the drop in pressure is so arranged that an equal number of heat units will be given up per stage, which, as will be remembered, does not correspond by any means to an equal drop in pressure.

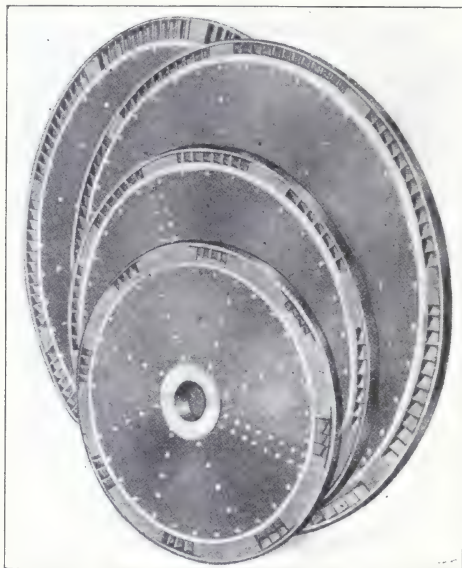


Fig. 51. Group of Diaphragms of Rateau Turbine.

In such a turbine as this, a foreign substance is not likely to injure more than one wheel, for it cannot pass the diaphragm separating the different chambers except through the nozzles, and, as there are many stages to such a turbine as this, the machine might run fairly well, even if one or two wheels were removed. It would, of course, give less power and poorer steam economy. The clearance between the nozzle and vanes should be small to prevent mingling of the steam jet with the stagnant steam in the casing, but

the clearance over the ends of the vanes is of little consequence, especially if a shrouding is used, for there is no tendency for steam to leak by the vanes, the pressure being constant throughout the chamber.

Rateau Turbine. The turbine using pressure stages only is best exemplified by the Rateau turbine, designed and developed by Professor Rateau, of Paris. His turbine is a horizontal, multi-stage, impulse machine and consists of sometimes as many as forty pressure stages, but usually less. Those built by the Western Electric Co. have from four to eighteen, depending upon the power and range of steam pressures. The large number of stages employed in these turbines means but little drop in pressure from stage to stage. Hence, the DeLaval type of expanding nozzle is not needed, it having been shown in the discussion of nozzles* that, if the final pressure is more than about 58% of the initial pressure, a parallel-sided passage or a slightly converging nozzle is sufficient to permit the proper expansion of the steam, and to secure the maximum available energy. In the Rateau turbine, the drop in pressure from stage to stage is much less than the limiting amount mentioned.

Diaphragm. As the steam passes through the turbine, it expands from one stage to the next and, of course, requires larger passage areas in each succeeding diaphragm. This is accomplished in general by increasing the number of openings rather than by increasing the size of them. The guide passages are arranged in groups, the number in each group being increased, and, consequently, the width of the group widened through successive stages until the openings finally extend entirely around the disk. To provide larger passage areas to

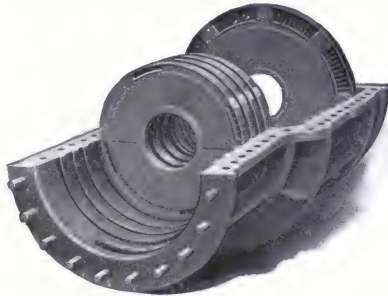


Fig. 52. Lower Half of Turbine Casing Showing Diaphragms in Place, and the Circumferential Grooves for Holding Them. *Western Electric Co.*

*Page 19, Part I.

take care of still further expansion of the steam, the diameter of the wheel and diaphragm must be increased, and, at the same time, the size

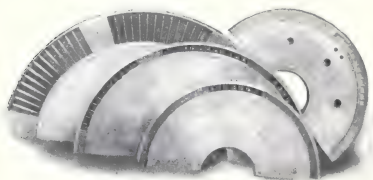


Fig. 53. Two-Piece Stationary Diaphragms with Distributing Vanes. *Western Electric Co.*

of the passage openings enlarged. In condensing turbines there are usually three diameters of wheel. In non-condensing turbines, two is generally sufficient, and sometimes only one is used. The

nozzles, or distributors, when only a portion of the wheel is open to steam admission, are set to have an angular advance on the preceding group, this advance being proportioned to the speed of the wheels, so that the steam jet as it leaves the revolving vanes will strike the next nozzle directly, avoiding any shock of impact against the solid wall of the casing. Any kinetic energy in the steam as it leaves the revolving vanes of one wheel is, therefore, directly available for use in the next stage. If the steam were brought up sharply against the solid casing wall, this residual energy would be lost to useful work, and there would result a further loss due to eddying of the steam in this particular

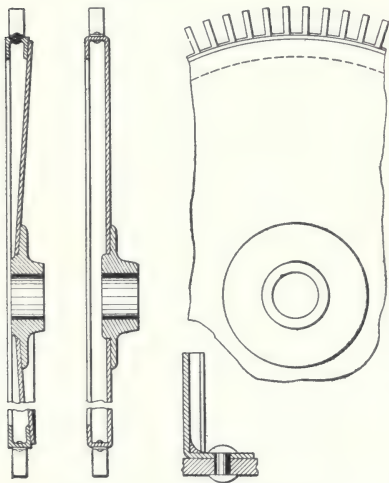


Fig. 54. Construction of Wheels of the Rateau Turbine.

cell. Fig. 51 shows a group of diaphragms of various sizes for this turbine, and the idea of increasing the extent of each succeeding group of nozzles.

There is a distinct advantage in this partial admission at the higher pressures, for, if the admission took place around the entire periphery of the wheel, the height of vane would necessarily be so small that the friction would be excessive. By using partial admission only, the vanes in these stages may be of much greater height than otherwise, and a few high vanes afford the same passage area for the steam that a large number of low vanes would offer, with, consequently, less friction.

Casing. Circumferential grooves are turned in the inside of the casing to hold the diaphragms in place, as shown in Fig. 52. The larger diaphragms are usually made in two pieces, as shown in Fig. 53. In all of them, the shaft passes through collars of anti-friction metal with clearances as small

as possible in order to prevent leakage from stage to stage. These collars are frequently provided with a labyrinth packing which will be explained later.

Vanes. The vanes are of crescent shape, similar in cross-section to those of the DeLaval turbine. The earlier ones had no protection at the outer ends, but the later ones have been provided with a shroud ring to give additional stiffness. The vanes and the shroud ring are generally made of nickel-steel as this is well adapted to resist corrosion.

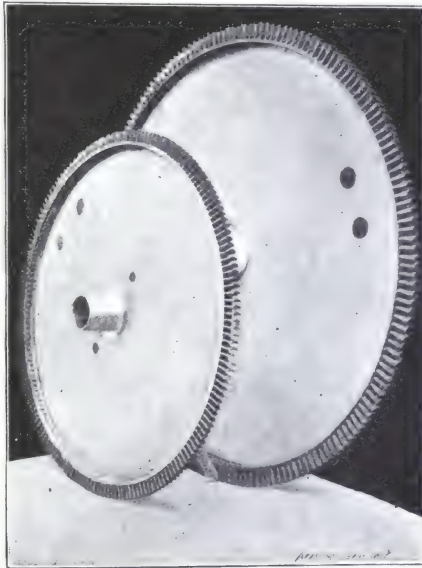


Fig. 55. Wheels of Rateau Turbine, Balanced by Holes in Disk.

Wheels. The rotating wheels are usually made of two plate-steel disks. One is flanged at the outer edge to which the vanes are riveted; the other, usually slightly conical, is riveted to the flange of the first disk, and both are riveted to the cast-steel hub. Fig. 54 shows the construction of this type of disk. The conical built-up disk makes a stronger wheel, but a flat disk is sometimes used. Some wheels are turned out of solid steel increasing in section toward the center to give greater strength. Each wheel is carefully balanced by itself on knife edges by drilling holes in the disk, the latter clearly showing in Fig. 55. When these wheels are assembled on the shaft

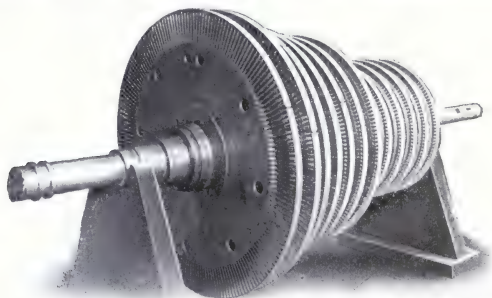


Fig. 56. Shaft and Wheels of High-Pressure Rateau Turbine.
Western Electric Co.

there is little likelihood that the complete rotor will be out of balance. Fig. 56 shows the wheels assembled on the shaft.

By-Pass Valve. This turbine is provided with a by-pass valve to carry overloads. This admits high-pressure steam into the intermediate stages and, although not permitting a complete expansion of such steam, it is an effective means of taking care of large overloads.

Bearings. The bearings are of the plain ring-oiling type, usually provided with water jackets. The shaft not being unduly long, there is little danger of whipping, and the speeds of rotation not being very high, special precautions are not necessary. Sometimes the turbines are supported by three bearings, as shown in Fig. 57, the high and intermediate stages being separated from the low by a third bearing.

Fig. 58 shows in section the turbine as built at Hawthorne, Ill., by the Western Electric Co., which is licensed to build the Rateau

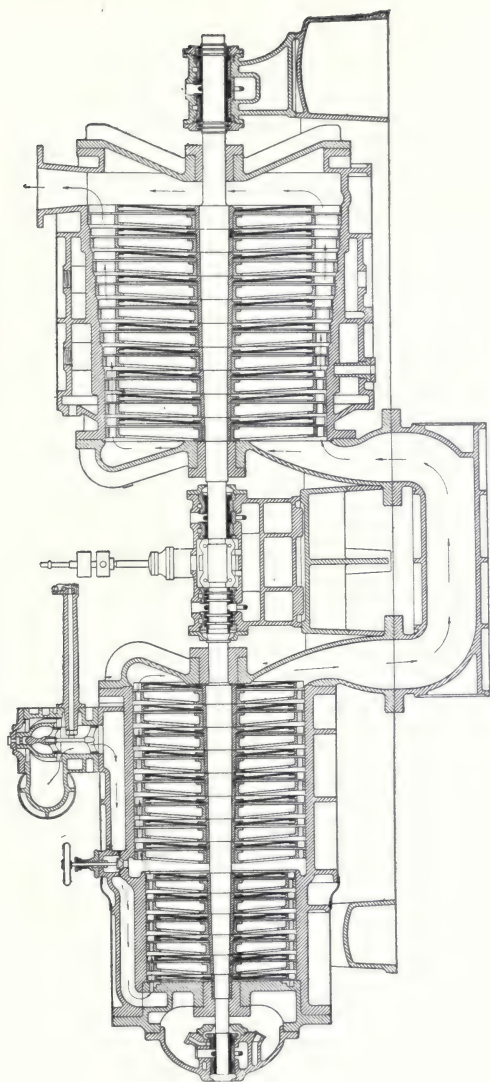


FIG. 57. Three-Bearing Support of Rateau Turbine.

turbines. Professor Rateau claims for a 1500-H. P. turbine at 1500 revolutions per minute only $1\frac{1}{2}\%$ leakage and bearing loss, and a $2\frac{1}{2}\%$ loss in friction of the wheels against the steam.

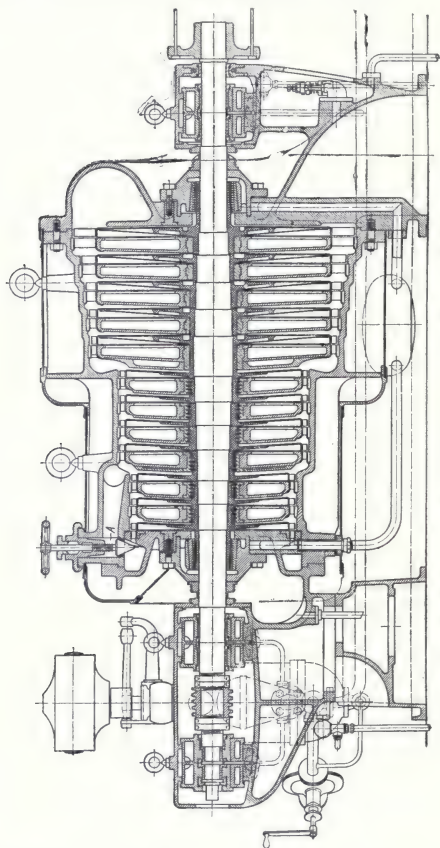


Fig. 58. Cross-Section of Rateau Turbine as Built by Western Electric Company.

Zoelly Turbine. The Zoelly turbine has been developed rather extensively abroad, and is being manufactured largely through a syndicate of builders including some American firms. It is a turbine essentially of the Rateau type, that is, a multi-pressure-stage turbine,

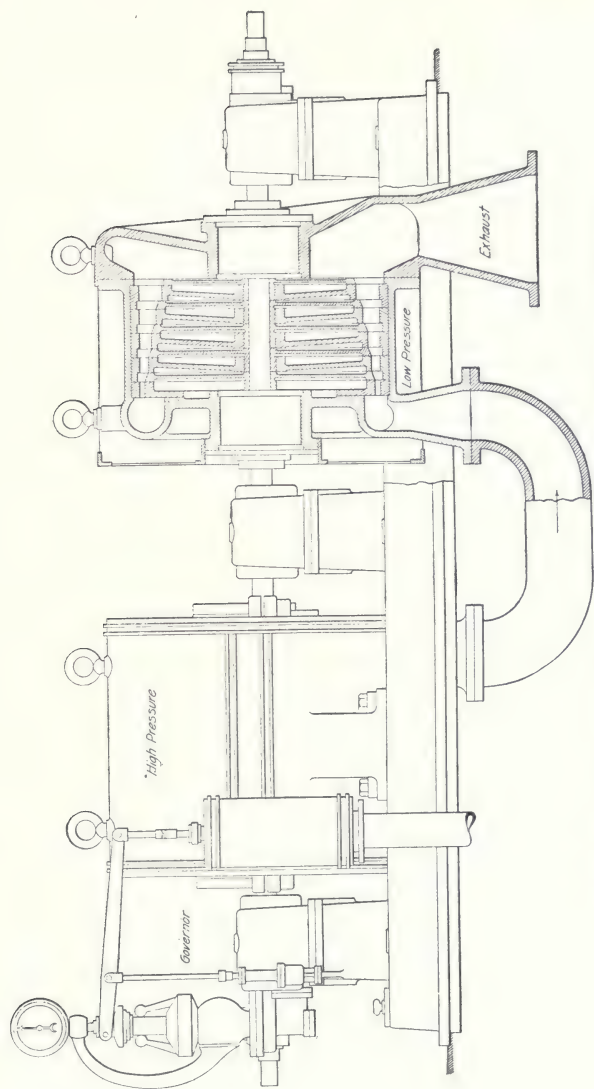


Fig. 53. Zoelly Condensing Turbine Showing Third Bearing between High and Low Pressure.

but it has fewer stages, usually not over ten for condensing and five for non-condensing, and it differs from the Rateau materially in detail. The blades are very much longer, sometimes being as much as half the diameter of the rotating wheel and, because of the fewer stages, many essential details are different. The turbine is sometimes divided into two parts when built condensing, the high pressure being separated from the low sufficiently for a third bearing to be placed between, as shown in Fig. 59.

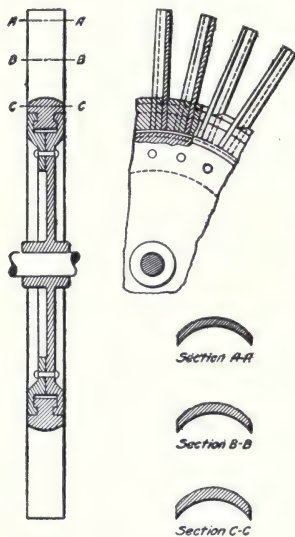


Fig. 60. Detail of Wheel and Vanes of Zoelly Turbine.

Vanes. There being fewer stages than in the Rateau, the steam velocities must be much greater, and, consequently, if the rotative speeds are to be the same, the diameter of the wheels must be greater. Exceedingly long vanes are used, which permit of a relatively small wheel disk; the centrifugal stresses in its rim, therefore, will not be materially greater than in other turbines. Again, the vanes are few in number compared with other turbines and are made as light as possible, tapering at the outer end in order to further reduce the centrifugal forces. The expansion from stage to stage is not enough to require diverging expansion nozzles except, perhaps, in the

high-pressure end, but the expansion is, of course, very much greater than in the Rateau turbine. In the latter turbine, the roots of the vanes are cut off parallel with the shaft, but in the Zoelly they are cut on a slope, giving a larger outlet than inlet to the vane. It might appear that this was done to permit of steam expansion in the vanes, but this is not so. Expansion is complete in the nozzles just as in the Rateau turbine, but, there being a much greater expansion from stage to stage, it follows that the area of the steam passages through each successive diaphragm must be greatly increased; this difference is

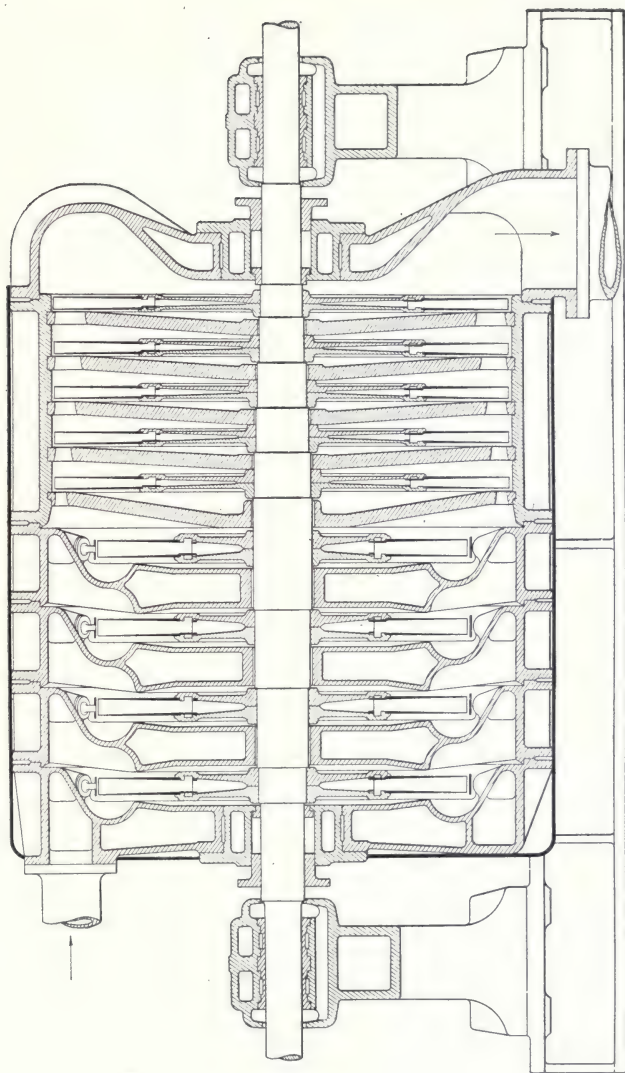


Fig. 61. Section Through One Type of Zoelly Turbine.

made up by increasing the depth of the openings in the succeeding diaphragm. Therefore, to permit a free passage of steam from an opening having one depth to an opening having a greater depth, and to prevent the formation of eddies, it is necessary to slope the root of the vanes. The vanes of the turbines being very much longer than in

the Rateau, it is quite feasible to increase the depth of the steam passages through the diaphragms.

Wheel. The vanes are set in slots cut in the rim of the wheel, and are secured by a clamp ring securely riveted to the main portion of the wheel, as shown in Fig. 60. It is probable that the excessively long vanes produce a considerably greater friction loss, revolving, as they do, in the steam-filled cells of the casing; but as there are comparatively few cells and, therefore, comparatively few revolving wheels, it is probable that this friction may not be any greater than would be expected in the Rateau

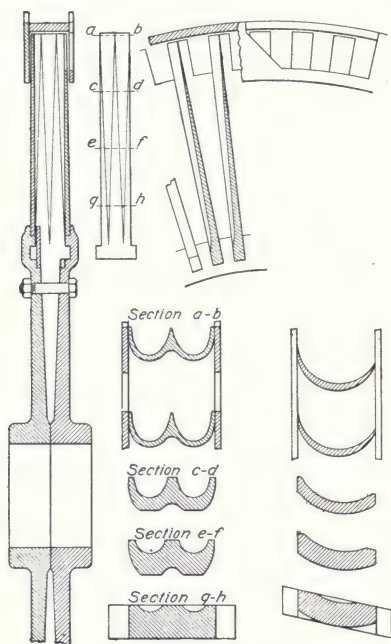


Fig. 62. Details of Construction of the Double-U-Shaped Vanes of the Zoelly Turbine.

turbine. Economies, as shown by test, do not appear to be essentially different from those of other first-class turbines. Zoelly turbines have been built as shown in Fig. 61. The typical style of long vane prevails, but, in the high-pressure stages, the vane is of the double-U-shaped cross-section, a detail of which is shown in Fig. 62. The steam jet necessarily impinges upon these vanes tangentially, instead of from the side.

Hamilton-Holzwarth Turbine. The Hamilton-Holzwarth steam turbine is another turbine of the Rateau type, the chief difference being that, instead of having the admission guides arranged in groups, the admission of steam in the high-pressure end takes place around the entire circumference of the diaphragm. The wheels of this turbine, therefore, would be smaller at the high-pressure end than would be the case with the Rateau turbine, and the vanes would be of appreciably

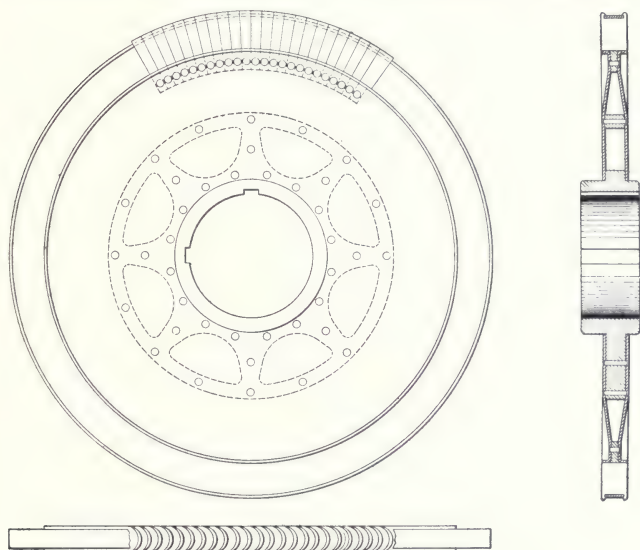


Fig. 63. Details of Wheel of Hamilton-Holzwarth Turbine.

less depth. Theoretically, the diameter of each succeeding wheel should increase as the steam expands but, for simplicity of manufacture, it is better to keep a number of wheels of the same diameter, increasing the length of blades to give larger passage areas. When the point is reached where this is no longer practical, the diameter of the wheel may be increased considerably, and the depth of the blade reduced.

There are approximately the same number of stages in the Hamilton-Holzwarth as in the Rateau turbine, and the speeds of revolution are not materially different. The running wheel shown in

Fig. 63 consists of a steel disk riveted to each side of the steel hub, or spider, which is keyed to the shaft. The outer edges of the disk are flanged outward, leaving a space to take the shank of the vane. The vanes are of drop-forged steel of the usual crescent-shaped cross-section. The shank of the vane fitted between two flanged disks, is riveted in place. Since the steam is admitted around the whole circumference of the diaphragm, vanes can be used better than nozzles to give the necessary expansion, and direct the steam upon the running wheels. Vanes could be used in the Rateau diaphragm, even with partial admission, but in such a case it has seemed simpler to drill openings through the diaphragm.

For small pressure drops, the vanes may be parallel, top and bottom, but, at the low-pressure end of the turbine where the volumes increase rapidly, they are usually deeper at the outlet than at the inlet, thus forming an easy passage of the steam to the next larger set of vanes.

A diaphragm, Fig. 64, separating the various cells, consists of a cast-iron disk bored loosely to fit the shaft. This disk has a groove *C* into which the shanks of the guide vanes are set, and a rivet holds them in place. A steel band is then shrunk over the ends of these guide vanes and this band projects into grooves in the casing to hold the diaphragm in place.

The Hamilton-Holzwarth turbine is manufactured by the Hooven, Owens, Rentschler Co., of Hamilton, O. Few machines, however, have been actually installed.

The Wilkinson Turbine. James Wilkinson has been developing a turbine of the multi-cellular impulse type with comparatively few stages, that is noteworthy because of the packing employed to prevent leakage of steam from stage to stage. It must be borne in mind that in a turbine of the Rateau type there are so many stages that there is comparatively little difference in pressure between one stage and the next, and, consequently, there is little tendency for steam to leak through between the shaft and the bushing in the diaphragm. As the number of stages is reduced, the difference in pressure is increased, and it then becomes essential that some sort of packing should be provided; otherwise the leakage is likely to be excessive. In practically all turbines, to prevent leakage of steam from the high-pressure end into the air, a *labyrinth* packing is used, consisting of a series of

grooves into which metal spring rings are fitted with slight clearances. These rings are not very different from the spring rings employed to pack the piston of the reciprocating engine, except that there are usually very many more of them. For steam to leak out, it must follow a tortuous course between the rings and sides and

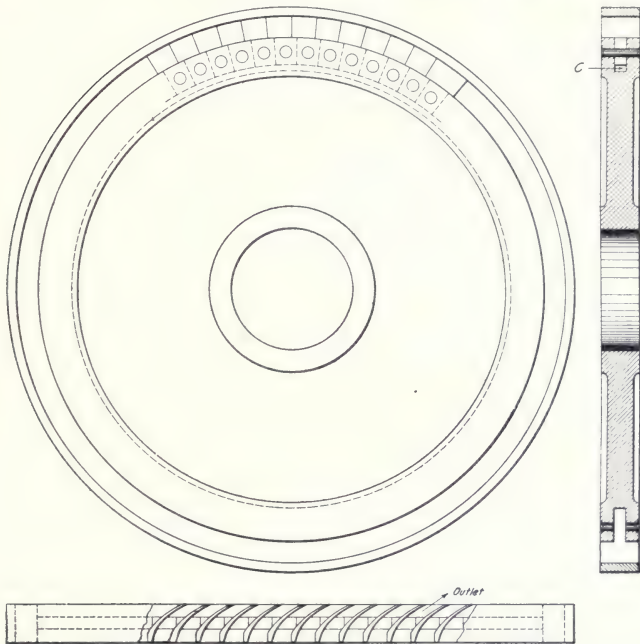


Fig. 64. Details of Diaphragm of Hamilton-Holzwarth Turbine.

bottom of the grooves, greatly expanding and condensing as it leaks through. The volume at the outer end is so large, due to this expansion, that the leakage becomes very slow. Condensation from such a packing is usually caught by a drip and taken to the hot well.

The Wilkinson idea is to groove passages along the shaft between certain rings of the labyrinth packing and the diaphragm bushings, as shown in Fig. 65. These grooves permit steam to pass to the

diaphragm bushing from a groove in the labyrinth packing, which is at slightly higher pressure than that in the cell at either side of the diaphragm bushing. Wet steam will, therefore, leak from the labyrinth packing to the diaphragm bushing and, as the pressure of the steam in the cells is less than the pressure of that which enters the bushing, this wet steam will expand, thereby vaporizing a portion of its moisture, and will then leak through into the cells of the turbine and have a possible chance of doing some useful work in the lower stages. The leakage from the labyrinth packing will be somewhat augmented by this means, but the leakage from stage to stage will be practically eliminated.

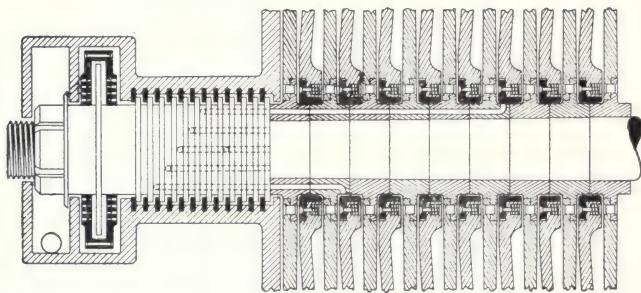
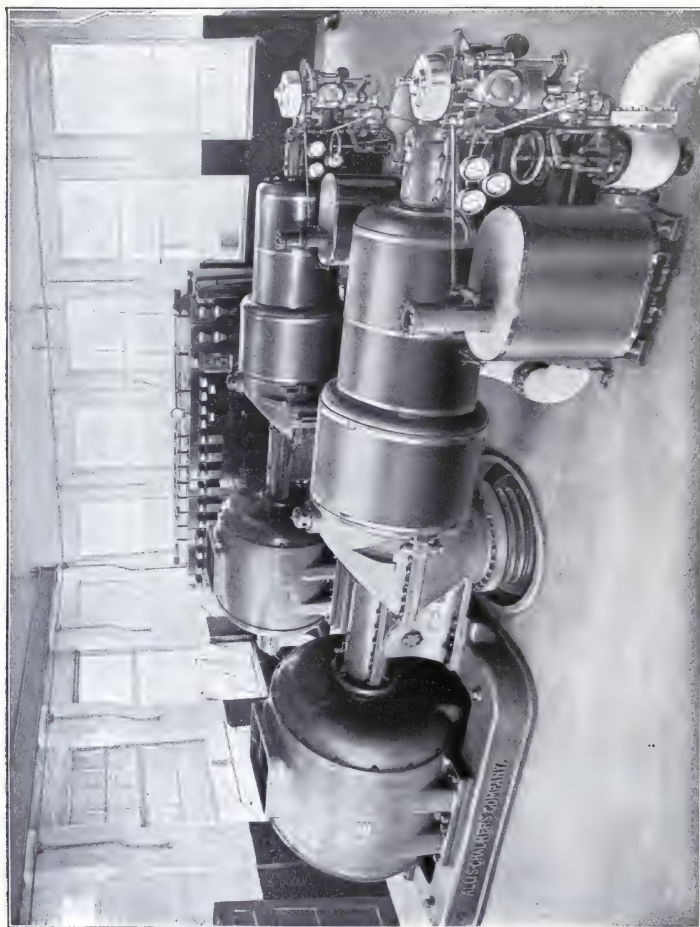


Fig. 65. Grooved Steam Passages and Labyrinth Packing of Wilkinson Turbine.

The Wilkinson turbine has a unique governing device, but is essentially of the Rateau type in principle, differing only in detail.

Kerr Turbine. The turbine built and developed by the Kerr Steam Turbine Company, of Wellsville, N. Y., is of the multi-stage type and differs from the Rateau in three important particulars. There are comparatively few stages; the vanes and buckets of the rotating wheels are of the double cup-shape sort after the style of buckets of the Pelton water wheel; and the steam is directed onto the wheel through nozzles instead of guide passages, striking the wheel tangentially in the plane of rotation instead of at the side. Otherwise the difference between this turbine and others of the multi-stage type is one of detail of construction.

The turbines are built in standard sizes with wheels 12, 18, 24, and 36 inches in diameter, with one to eight stages, developing up to



TWO 1,500-K. W. ALLIS-CHALMERS STEAM TURBINES AND GENERATORS IN PLANT OF THE KERR MILLS
OF THE AMERICAN THREAD CO., FALL RIVER, MASS.
Allis-Chalmers Co., Milwaukee, Wis.

600 H. P. The power developed in a single wheel of given diameter will naturally depend upon the number of nozzles in action.

The buckets are made of drop forgings and are reamed out with a special reamer, each cup being, in cross-section, a surface of revolution. Fig. 66 illustrates this type of bucket.

The wheel is a solid disk and the buckets may be attached by riveting, as shown in Fig. 66, or by dove-tailing, as shown in the same figure, the latter being the preferred and more usual form of construction. Each wheel is carefully balanced and then assembled on the shaft, as shown in Fig. 67.

The nozzles are of machine steel, screwed into a steel nozzle body and this latter securely riveted to the diaphragm casing.

Fig. 68 shows the nozzles, wheel, and diaphragm.

The shell is cast in sections, one for each stage. These are set between two end castings, turned, and bored. Tongue-and-groove joints on these castings insure correct alignment, and fibrous packing

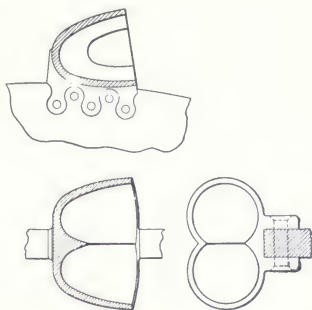


Fig. 66. Buckets of Kerr Turbine.



Fig. 67. Kerr Turbine Wheels Assembled on Shaft.

in the grooves, in addition to metal contact of the surfaces, insures steam tightness. A casing built in this manner possesses two distinct advantages over a solid casing. There is no probability of cracking due to rapid temperature changes, and the size of turbine may be

increased by the addition of more units. The end castings carry the weight of the turbine, and have supports bolted to the bed plate.

The shaft, where it passes through the diaphragm, is fitted to a bronze bushing with a few thousandths of an inch clearance. This bushing seats on the metal surface of the diaphragm with latitude for slight side motion. It is kept to its seat by the steam pressure, but can move sideways to accommodate any whipping motion of the shaft. Fig. 69 shows the sectional view of the Kerr turbine.

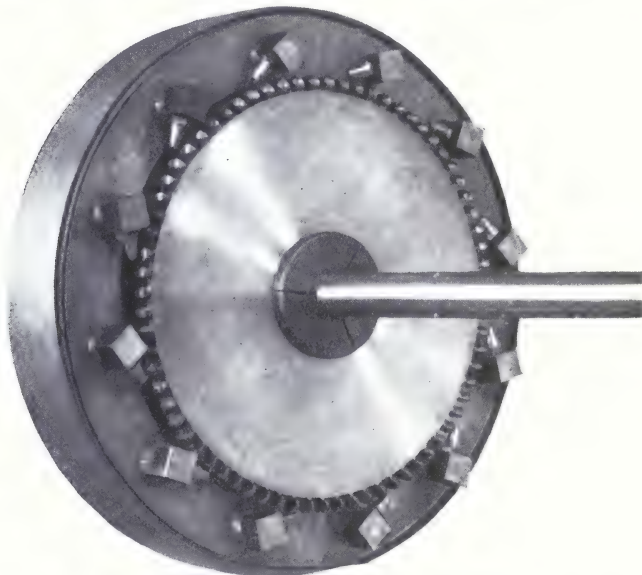


Fig. 68. Nozzles, Wheel, and Diaphragm of Kerr Turbine.

COMPOUND IMPULSE TURBINE WITH PRESSURE STAGES AND VELOCITY STEPS

One of the simplest and most effective ways of compounding turbines is by both pressure stages and velocity steps. The turbine shell is divided by diaphragms into a number of different cells, seldom more than five, except for marine work, where more are necessarily employed. There being comparatively few stages, it will, in general,

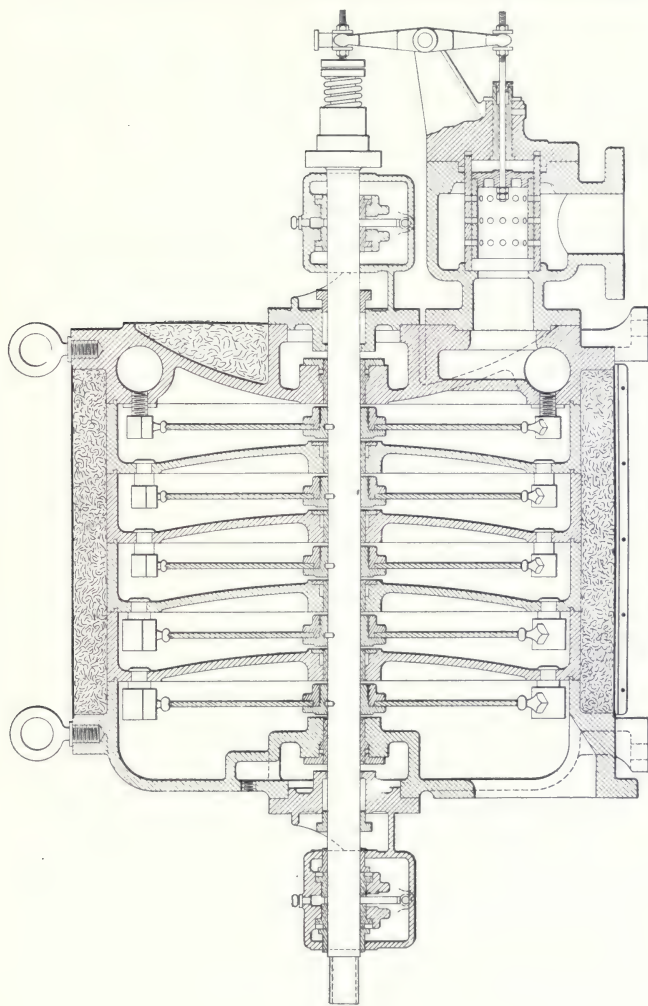


Fig. 69. Sectional View of Kerr Turbine.

be necessary to employ diverging nozzles so proportioned that the steam may be completely expanded within the confines of these nozzles, from the initial pressure to the pressure in the chamber into which the stream is discharging. It will be remembered that multi-stage turbines of the Rateau type do not require expanding nozzles because of the relatively small drop in pressure from stage to stage.

Turbines of the type now being described differ from the Rateau type in another most important particular. Each cell, or chamber, of this type of turbine contains two or more sets of rotating vanes, while turbines of the Rateau type have but one wheel and one set of vanes in each chamber. The steam leaving the nozzles impinges upon the first set of running vanes, and, as the steam leaves these vanes, it flows into a set of guides of some sort, and is returned to other vanes of the same set, or to a different set of vanes on the same wheel, or to the vanes of a separate wheel, as the case may be. The steam may pass through from one to three sets of redirecting guides, and may impinge upon two to four sets of rotating elements. It is immaterial, so far as the principle of the action is concerned, whether the steam acts successively upon a number of rotating wheels or whether it is returned again and again into different vanes of the same wheel. If the latter form of construction is adopted, the turbine will necessarily be more compact and the rotating shaft will be shorter.

The more velocity steps the turbine has per stage, the less number of stages will be necessary, but, in general, it is found more economical to increase the pressure stages than to increase the velocity steps. It must be remembered that in this type of turbine the steam is completely expanded within the nozzle, and that the temperature and pressure of this expanded steam are the same within the confines of any particular cell. As the steam passes through successive rotating vanes, it gradually loses velocity and, consequently, the succeeding passages must be made larger and larger, in order that the same volume of steam may pass through at the lower velocity in a given time. In other words, the passage area must increase in proportion to the reduction in velocity. If this point is not clearly borne in mind when looking at the vanes of a turbine of this type, one might think that the increased size of passages were to provide for increased expansion of the steam. In the Terry steam turbine, which is of this

type, a portion of the steam is allowed to escape as it passes through successive buckets, so that the volume is gradually reduced as the velocity decreases.

Curtis Turbine. Undoubtedly the best-known turbine of this type is the Curtis, which has been developed by the General Electric Co., and is being built at their works at Schenectady, N. Y., and

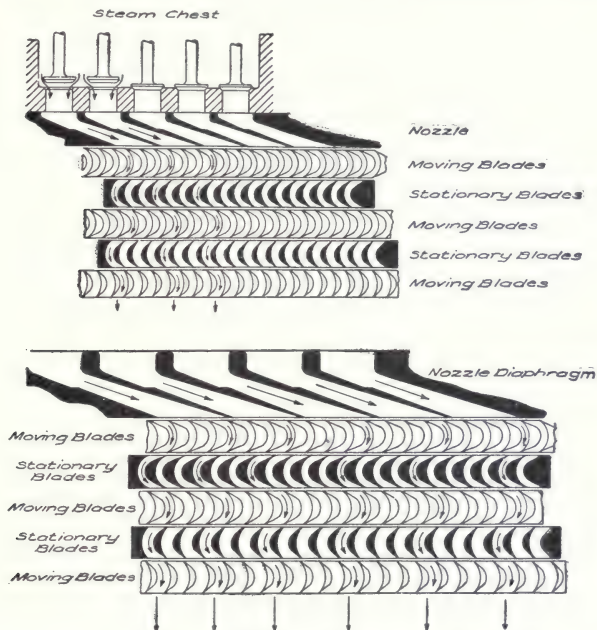


Fig. 70. Diagram of Nozzles and Buckets in Curtis Steam Turbine.

Lynn, Mass. Rights to build the Curtis turbine for marine propulsion are controlled by the Fore River Ship-Building Co., of Quincy, Mass.

For ordinary purposes up to 9000 kilowatts, the Curtis turbine does not have over four stages, with two velocity wheels per stage, except in the larger sizes of turbine, when a fifth cell is provided, which contains a single rotating wheel without redirecting guides,

for the purpose of abstracting any residual velocity there may be in the steam after passing the fourth stage. The marine turbine of this type installed in the U. S. Cruiser *Salem* had seven pressure stages

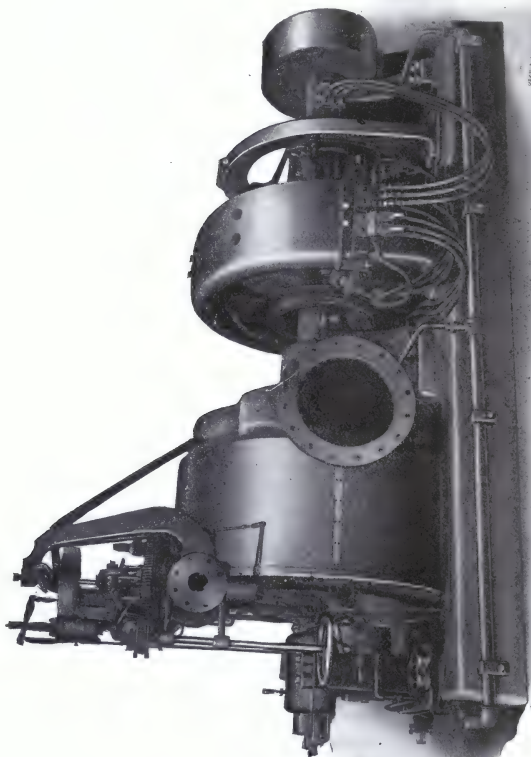


Fig. 71. 300-K. W. Curtis Turbine Connected to Direct-Current Generator.
General Electric Co.

with four velocity steps in the first stage, and three in each of the others.

Fig. 70 shows in diagrammatic form the principle of the steam action in the Curtis turbine. Steam passes from the steam chest

through the nozzles, each of which may be closed by a valve operated from the governor. The number open at any one time depends upon the load. There are enough valves and nozzles to take a large overload without the use of the by-pass which would admit high pressure steam into the lower stages. Such a by-pass is, however, usually provided. It works automatically and admits high-pressure steam to a set of auxiliary nozzles fitted into the second stage of the turbine, thus increasing the power with but a slight sacrifice in steam expansion and consequent economy.

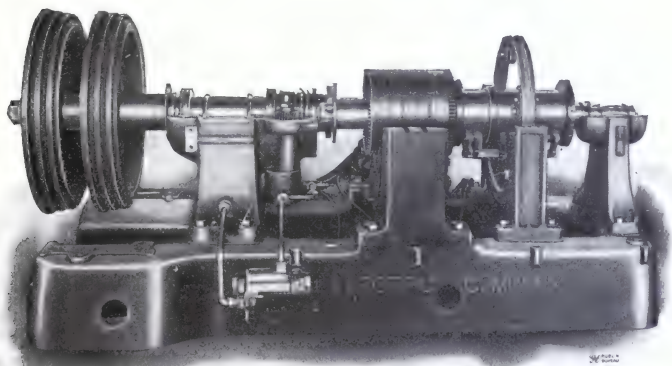


Fig. 72. 100-K. W. Curtis Turbine and Direct-Current Generator Partly Assembled.
General Electric Co.

The nozzles are designed with a view to producing such a drop in pressure from stage to stage that equal amounts of work are done in each stage. This does not correspond to an equal drop in pressure by any means, for there are more heat units in a given drop of pressure in the lower than in the higher ranges. With many stages in a Curtis turbine, the low-pressure diaphragms might be fitted with plain cylindrical nozzles.

Steam enters the first row of rotating vanes* from the nozzles,

*The rotating vanes are called *buckets* by the General Electric Company.

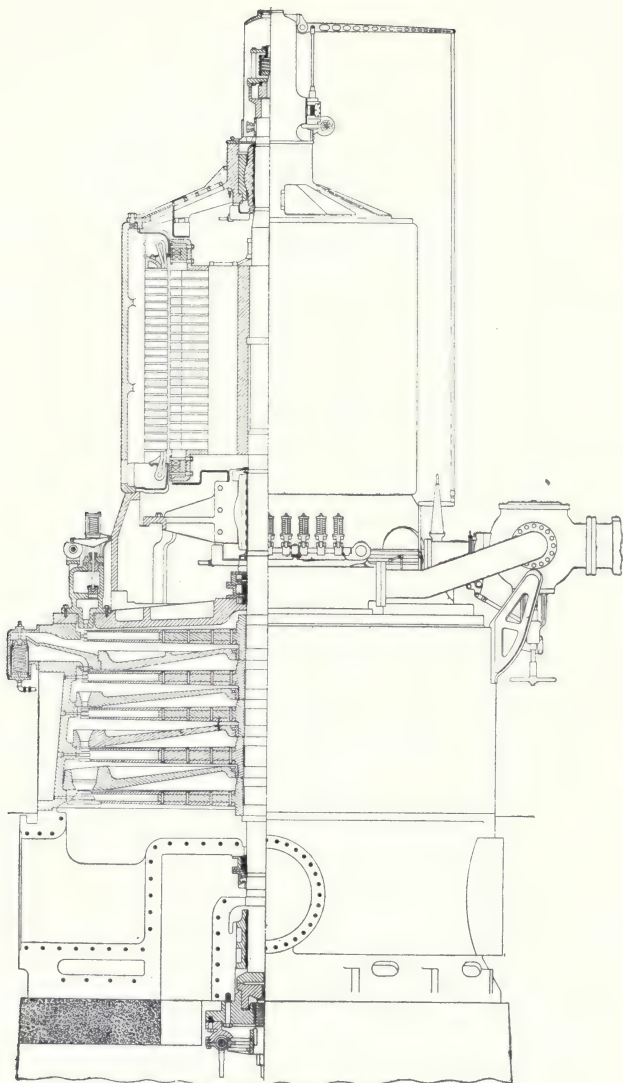


Fig. 73. Vertical Curtis Turbine of 9000-K. W. Capacity.

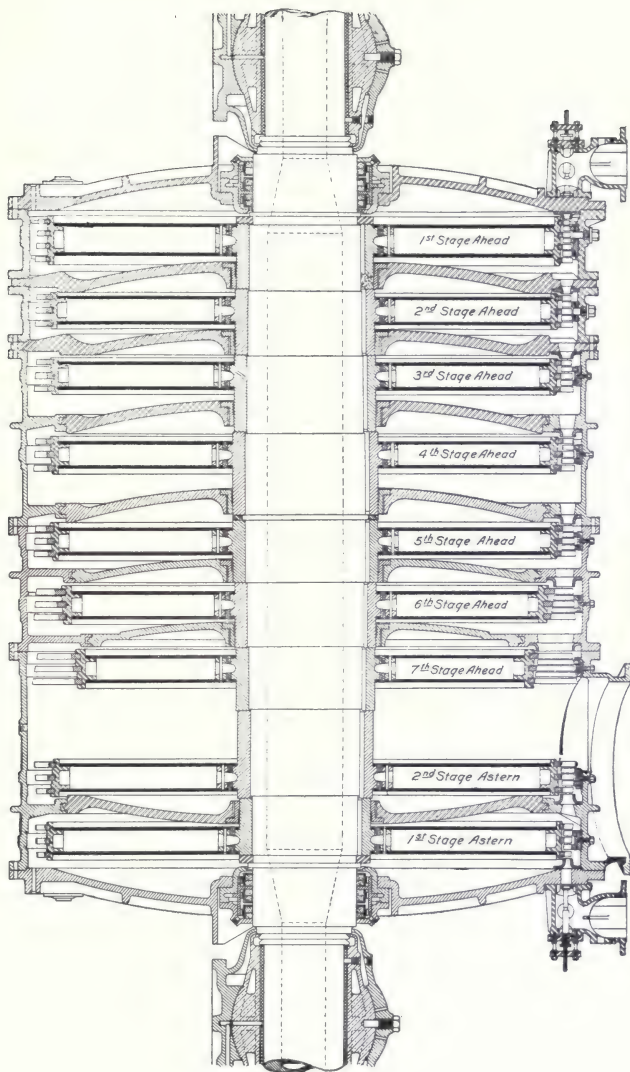


Fig. 74. Sectional View of the Curtis Turbine in the U. S. S. *Salem*.

is deflected from these vanes to the guides or *intermediates*, as they are technically called by the General Electric Company, and is re-directed to the next row of moving vanes, and so on, passing from the last row directly into the next set of nozzles, and again to the next stage as before.

As has already been mentioned, the number of pressure stages and velocity steps in the Curtis turbine vary with the size of the unit. In sizes from 75 to 300 kilowatts, there are two stages and three velocity steps, as shown in the diagram, Fig. 70. The 500-kilowatt* to 3000-kilowatt sizes are four-stage with two velocity steps per stage, while those of over 3000 kilowatts are five-stage with but a single wheel in the fifth chamber and, of course, no redirecting buckets. The turbines, in sizes up to 300 kilowatts, are generally of the horizontal type; the larger sizes are vertical. Fig. 71 shows a 300-kilowatt horizontal turbine, Fig. 72, a 100-kilowatt turbine, partly assembled, and Fig. 73, a vertical turbine of 9000-kilowatt capacity. A sectional view of the turbine as fitted in the U. S. S. *Salem* is shown in Fig. 74.

It may be interesting to note the speeds of rotation of Curtis turbines of various sizes.

500 kilowatts	approx.	1800 R. P. M.
1000 kilowatts	approx.	1200 R. P. M.
2000 kilowatts	approx.	900 R. P. M.
5000 kilowatts	approx.	800 R. P. M.
9000 kilowatts	approx.	750 R. P. M.
Marine Turbines	approx.	250 R. P. M.

Casing. Fig. 75 shows the section of the casing of a four-stage turbine. It is built of cast iron of four parts for sizes up to 3000 kilowatts, and six parts for 5000 kilowatts and larger sizes. This casing holds the stationary reversing vanes and supports the diaphragm, the details of which are clearly shown in Fig. 75. *A* represents the inlet nozzle for the first stage, *B* the guides or intermediates, *C* the diaphragm separating the first and second stage, *D* the ledge on the casing which supports the diaphragm, *E* the spider of the rotating wheel, *F* the wheel plates, and *G* the distance piece at the outer rim of the wheels.

*The General Electric Company builds a special 500-kilowatt, vertical turbine having only two stages, with, consequently, three rows of revolving vanes per stage.

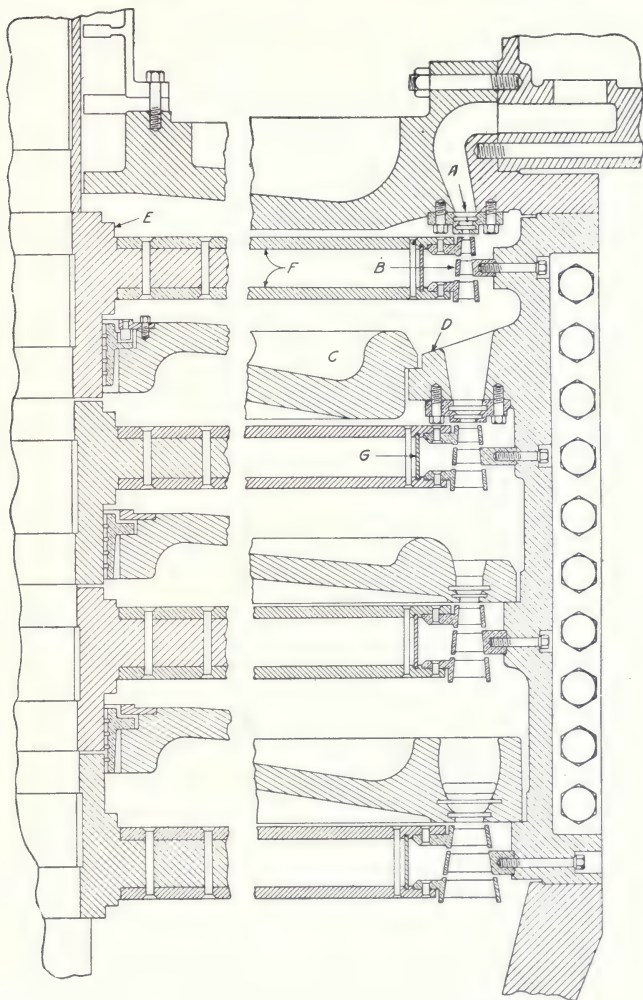


Fig. 75. Section of Casing of Four-Stage, Curtis Vertical Turbine.

Nozzles. The nozzles are grouped together, not as in the Rateau turbine, but in one single group for each stage, thus admitting a single steam belt to a part of the wheel periphery only. The Rateau turbine also admits steam

to only a portion of the periphery but, in this turbine, instead of one, there are several groups of nozzles arranged at equal intervals around the periphery.



Fig. 76. Group of Nozzles for High-Pressure End.

Fig. 76 shows a group of nozzles for the high-pressure end of the turbine. The outlet is of rectangular section, slightly rounded at the corners, which the makers claim gives better results with this type of turbine than the elliptical outlet used in the DeLaval. The nozzles are reamed out of bronze castings and riveted to the casing.

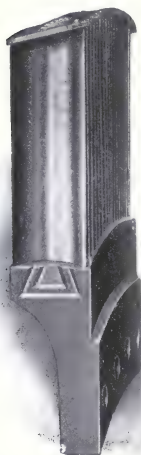


Fig. 78. The Dovetailed Type of Bucket in the General Electric Curtis Turbine.



Fig. 77. Group of Buckets for Curtis Turbine.

Vanes. The vanes, usually called *buckets* by the Curtis manufacturers, are crescent-shaped in cross-section, as is common for impulse turbines. In the smaller sizes of turbines, the buckets are cut by special machinery directly on the periphery of the wheel, which is made of cast steel. In some

of the larger sizes, the buckets are cut on the circumference of a ring, the inner diameter of which corresponds to the diameter of the rotating wheel. This ring is then riveted onto the wheel disk.

In the largest sizes the vanes are often milled from sections of cast bronze, some two feet in length, containing fifteen to twenty-four buckets each, Fig. 77, depending upon the size of the buckets, these segments then being riveted to the wheel. A metal shroud



Fig. 79. Revolving Wheel for 2000-K. W., Curtis Steam Turbine.
General Electric Co.

ring covers the outer end of the buckets, and is securely riveted to a projection on the bucket made for that purpose. Fig. 77 gives a view of a section of such buckets ready to rivet to the wheel.

It is becoming a general practice with the General Electric Company to make their buckets separate, and fasten each in place on the wheel or segment, by setting a dove-tailed piece at the lower end of the bucket into a corresponding groove cut in the segment casting, as in

Fig. 78. These buckets are correctly spaced by means of small distance pieces which are fitted alternately with the buckets.

Wheels. Fig. 79 shows the rotating wheels of a Curtis turbine, with two sets of vanes so spaced that, as the wheel rotates, the one set of vanes will be on each side of the guide vanes or *intermediates*. This is equivalent to having two rotating wheels, but in construction is much simpler. Each wheel, as shown, rotates in a chamber by itself. Fig. 79 and Fig. 80 show the old style of bucket segments which offer a very rough surface. In the present style, Fig. 77, the segments have no external webs and the riveting is flush. The wheels, except in the larger sizes, are made of solid steel securely keyed to the shaft at the hub. This hub varies in length according to the stage in which it revolves,



Fig. 80. Revolving Buckets for Curtis Steam Turbine. General Electric Co.

the low-pressure stages necessarily being wider than the high-pressure stages. The wheel gradually tapers toward its periphery, thus maintaining a section of approximately uniform strength. Wheels for the largest sizes of turbine are built up of a

cast-steel hub, or spider, to which are riveted steel disks, one on each side, and between them is riveted a cast-steel distance ring. The vanes are riveted in segments onto the outer edges of the disks, as is clearly shown in Fig. 73 and Fig. 74.

A channel is turned in the wheel disk near the rim to receive the bucket segments. By means of this channel, therefore, the rivet, which passes through the wheel disk and the bucket segments, is relieved of a portion of the strain produced by the centrifugal forces. Fig. 81 shows the rotor of the four-stage turbine, with wheels assembled on the shaft. The separate sets of vanes, one on each side of the wheel, and the channel in the rim may be clearly seen.

Guides. The guide vanes, or *intermediates*, are necessarily in one group of sufficient extent to catch the steam belt as it issues from the rotating wheels. The group naturally extends at first over but a small arc of the circumference, this arc increasing in extent in each stage until both nozzles and intermediates entirely surround wheel and

casing. The guide vanes are in cross-section like the revolving vanes, but are set in a reverse position. They are set with a certain angular advance from the nozzles, depending upon the speed of rotation, so

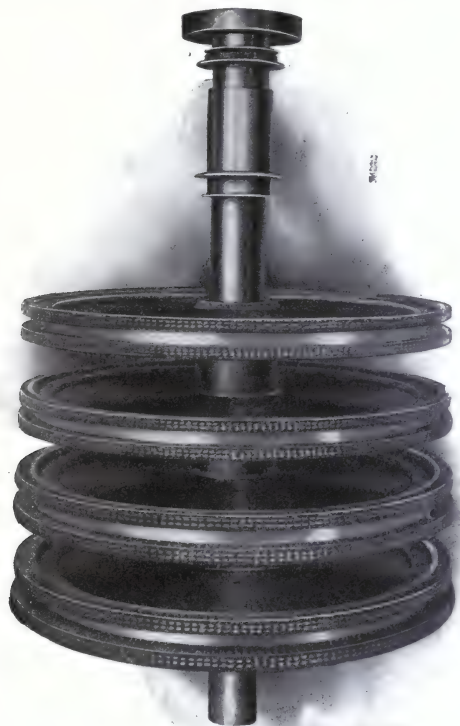


Fig. 81. Rotating Wheels of 500-K. W., Curtis Turbine. *General Electric Co.*

that the steam leaving the vanes will strike full upon them. Like the vanes, they have to provide a successively increasing passage area to allow for the lower velocity of the steam, if more than one set of guides is used per stage.

Diaphragm. The style of diaphragm is best shown in Fig. 74 and Fig. 75, which also clearly show the construction of the large wheels. The diaphragm is an iron casting—steel in the higher stages, because there the drop in pressure is greatest and consequently there is a greater load on the diaphragm—slightly dished in shape toward the high-pressure side to give greater strength. It is provided with bronze bushings where the shaft passes through, these bushings being fitted with slight clearance to prevent leakage. The ends of the casing are packed with carbon packing to prevent leakage of steam into the air at the high-pressure end, and leakage of air into the turbine at the low-pressure end, which leakage would have a detrimental effect upon the vacuum.

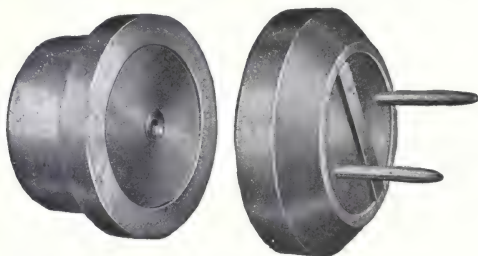
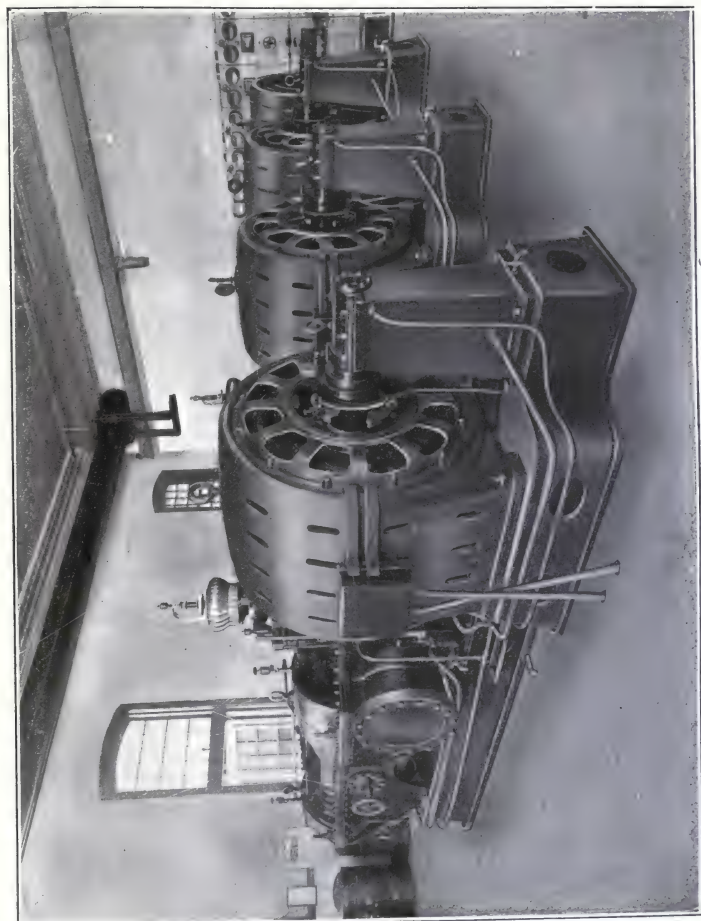


Fig. 82. Bearing Surfaces in Step Bearing of Curtis Turbine.
General Electric Co.

Bearings. All sizes of the Curtis turbine of 500 kilowatts and over are of the vertical type with but one working bearing, located at the lower end of the shaft. This bearing consists of a short cylindrical block of cast iron fitted with two dowel-pins and a key, as shown in Fig. 82, and a corresponding cast-iron block having a hole through its center, into which is threaded a pipe for supplying some form of lubricant, either water or oil.

Fig. 83 shows the bearing assembled on the lower end of the shaft. The upper bearing with dowel-pins and key fit into corresponding dowel holes and key-way in the bottom of the shaft, and rotate with it. When the oil is supplied to the bearing, which is, of course, under a high pressure, it fills the central circular space between the blocks and forces them slightly apart. The oil then escapes between the annular edges of these two blocks and is col-



FOUR 400 K.W. TURBINE GENERATING UNITS.
Westinghouse Electric and Manufacturing Co.

lected into a drain and returned to the original supply. If water is used for a lubricant, it is allowed to flow up into the base of the turbine and mingle with the exhaust steam on its way to the base of the condenser. The pressures maintained by the lubricating pump in practice vary from 180 to 450 pounds per square inch. It is thus seen that the two bearing blocks do not come into actual contact, but that the weight of the turbine is supported upon a film of lubricant. Should the lubricating pump fail in its supply, no more serious damage would occur than the abrasion of the step bearing, and a new one could readily be inserted, as the figure will show.

Riedler-Stumpf Turbine. The turbine developed by Professors Riedler and Stumpf for the larger powers necessitating lower speeds was provided with two to four pressure stages, with two velocity steps per stage, and this turbine expanded the steam on the same

principle as the Curtis, but the details of construction and general arrangement were entirely different. In the two-stage, and even in the four-stage, turbine, the overhung type of wheel developed in the single-stage turbine was adhered to, the generator being between the two turbine wheels with a bearing at each end. Fig. 84 shows this arrangement in a 5000-kilowatt, two-stage turbine revolving at 750 R. P. M., with two velocity steps to each stage. In this particular turbine, the steam is returned through U-shaped guide passages to a second set of buckets on the same wheel, these buckets being larger

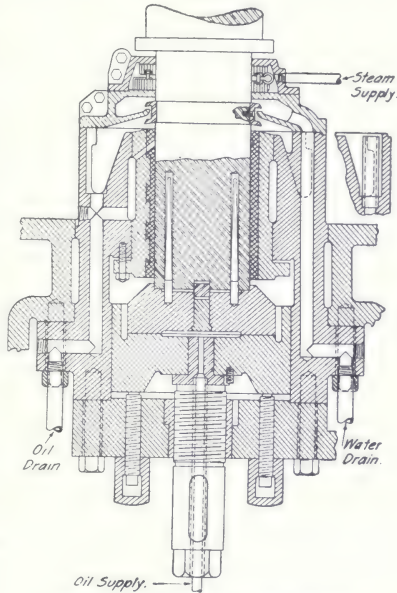


Fig. 83. Step Bearing of Curtis Turbine.

than the first because of the lower velocity. Fig. 85 shows a 500-kilowatt, four-stage turbine of the same type. These have been built both vertical and horizontal, the vertical arrangement resembling externally the Curtis turbine.

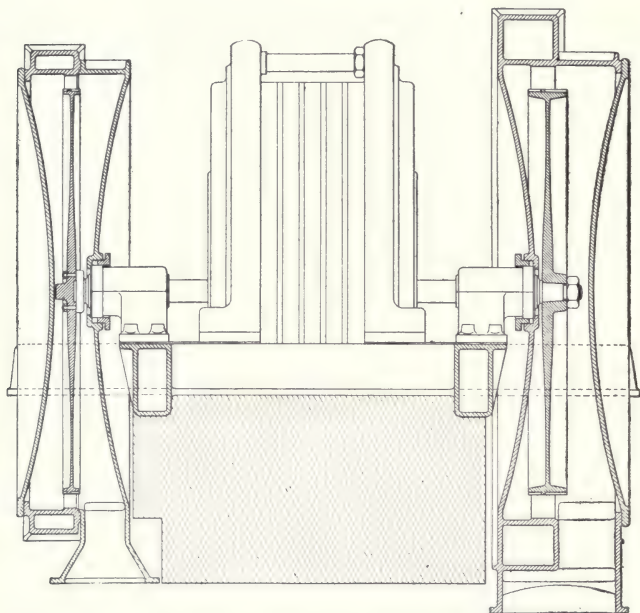


Fig. 84. Overhung Wheels of the Riedler-Stumpf Turbine.

Fig. 86 shows a vertical four-stage, two-step turbine of 750 R. P. M. developing the same power as that shown in Fig. 85.

The Riedler-Stumpf turbine was formerly manufactured by the *Allgemeine Electricitäts Gesellschaft* of Berlin, but, as this company is now licensed to manufacture under Curtis patents, it does not appear that the manufacture of the former type is being actively carried on at the present time.

Terry Turbine. In sizes developing over 300 H. P., the Terry Steam Turbine Co. are compelled to resort to the two-stage type.

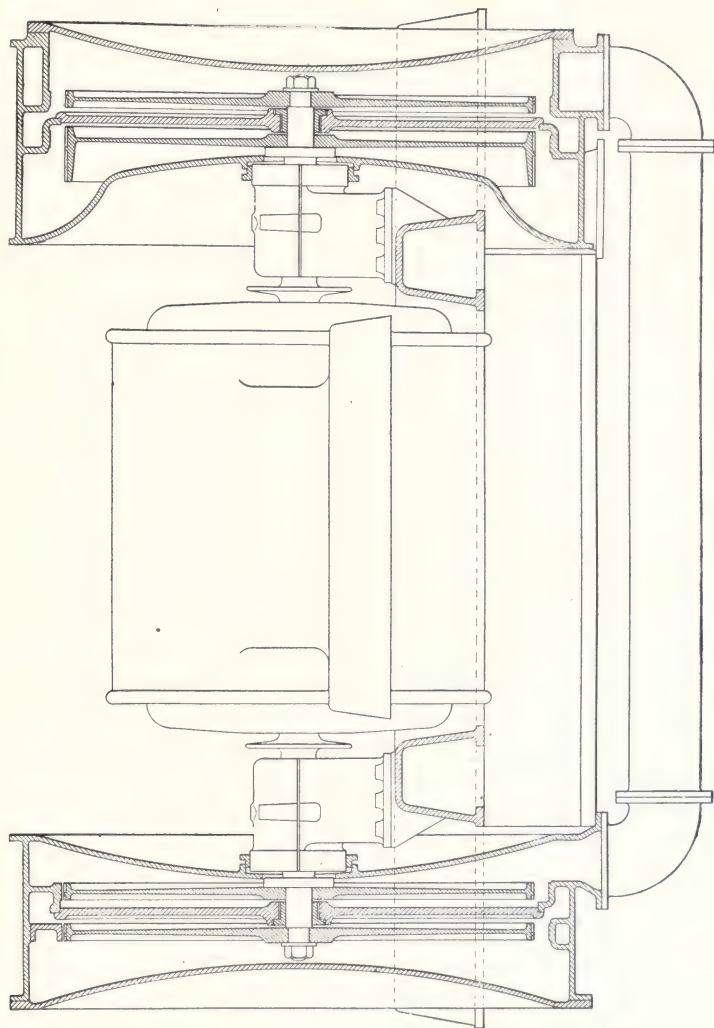


Fig. 85. 500-K. W., Four-Stage, Riedler-Stumpf Turbine.

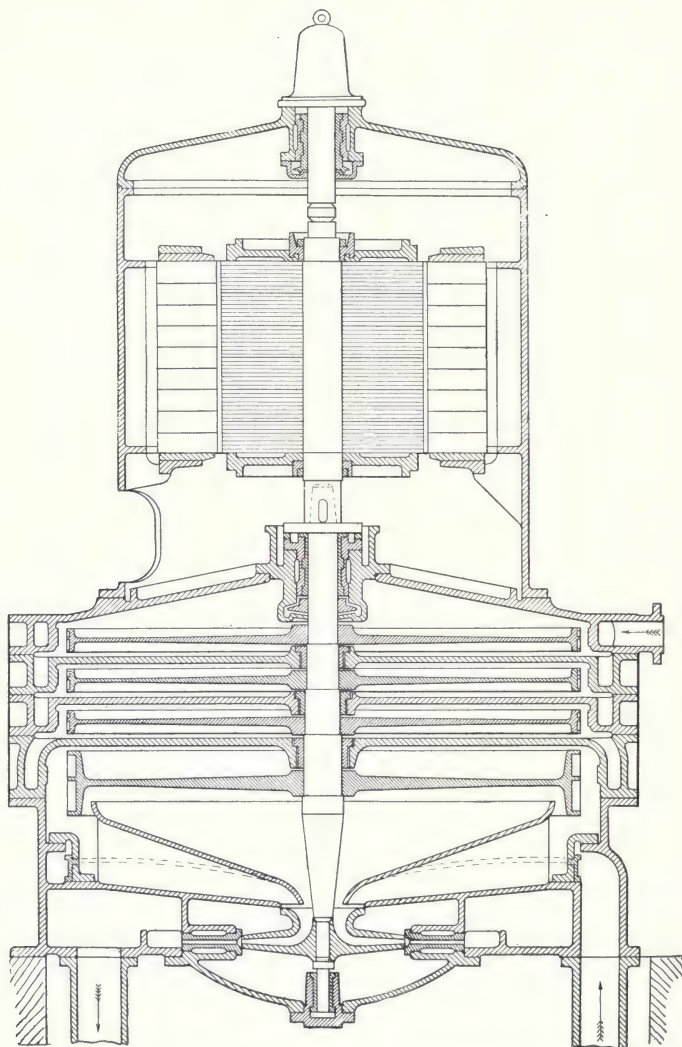


Fig. 86. Vertical, Four-Stage, 500-K. W., Riedler-Stumpf Turbine.

At present, they are not building turbines of over 600 H. P. but, in very large sizes, more pressure stages would be required. Their compound-pressure turbines are not essentially different from the single-stage in detail of construction. Fig. 87 shows a section of this turbine.

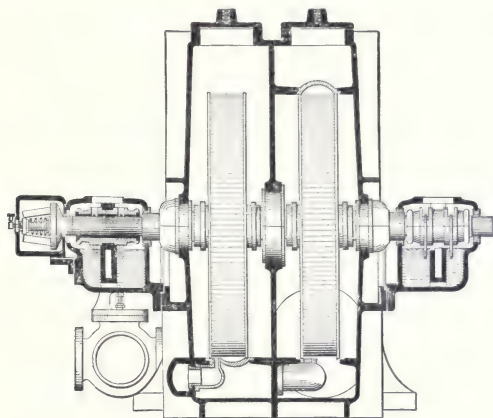


Fig. 87. Section Through Two-Stage, Condensing, Terry Steam Turbine.

REACTION TURBINES

In this type of turbine, the steam expansion takes place, not alone in the nozzles and guide passages, as in all the types of turbine previously described, but in the revolving vanes as well, approximately half the expansion taking place in each. In this type of turbine, the difference in pressure on the two sides of the vanes brings about a leakage of steam over the tips of both the rotating vanes and guide vanes. That this leakage may not be unduly large, the drop in pressure is made small from stage to stage. The leakage at the high-pressure end gradually expanding does some work in the succeeding stages and becomes relatively small at the low-pressure end. Clearances, however, in this type of turbine are all-important and, other things being equal, that reaction turbine showing the best economy will be the one with the smallest radial clearance. To make

the leakage in the high-pressure stages as small as possible, the vanes should be made as long as convenient, about five per cent of the diameter of the rotor being considered a minimum.

In turbines of the Rateau type, there is a diaphragm separating the different stages. This extends close to the shaft, permitting leakage only through the small annular space between shaft and bushing, a comparatively unimportant matter, as these clearances may be made very small. Besides this, carbon or labyrinth packing might be provided, which would make the leakage from stage to stage almost negligible. Moreover, there is no tendency to leak over the tips of the running vanes, because the pressure is the same on both sides of them.

In the reaction type of turbine, however, there is no diaphragm from stage to stage, and, instead of each rotating wheel being a separate element, it is customary to fasten the vanes in rows to the rotating drum or cylinder. In this style of machine, one set of guide vanes and one set of revolving vanes constitute a stage, and it can readily be seen that the opportunity for leakage between the tips of the stationary vanes and the drum of the rotor is very much greater than in the Rateau type, the annular space in the two types being proportional to the diameters of shafting and drum. The tendency to leak over the tips of the running vanes is even greater, because of a greater diameter. However, friction losses, which are approximately proportional to the square of the steam velocity with reference to the vanes, will be very much less in the reaction type of turbine than in the impulse type, because the steam velocities are comparatively low.

Parsons Turbine. The Honorable Charles A. Parsons, of England, is responsible for the successful development of the reaction type of turbine. His first turbine, made in 1884, was of 10 H. P., 18,000 R. P. M., and when running non-condensing with 92 lbs. of steam pressure by the gauge, it is claimed that but 25 lbs. of steam per brake horse-power per hour were consumed. In 1888, a 50-H. P. turbine at 7000 R. P. M. was constructed, and soon after a 200-H. P. at 4000 R. P. M. showed good economy. It must be remembered that the chief problem of the turbine designer has been to reduce rotative speeds without material sacrifice in economy.

Parsons turbines are manufactured in the United States by the

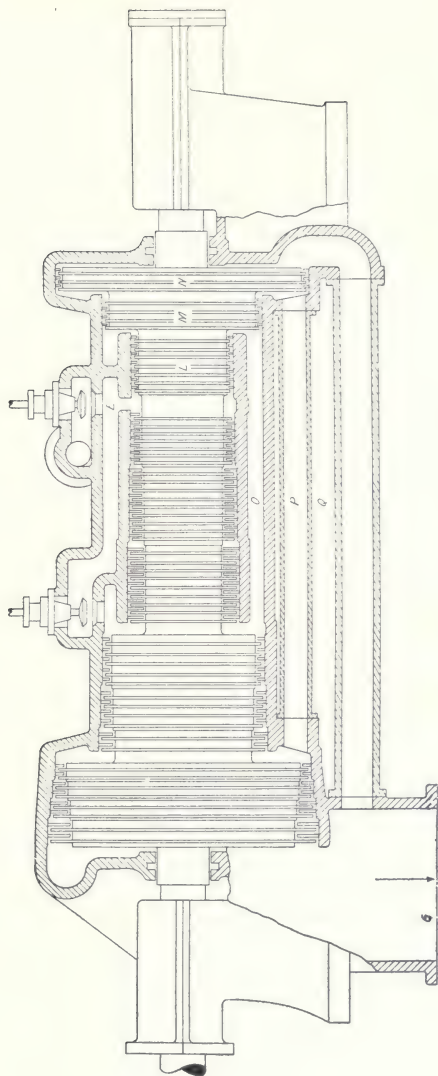


Fig. 65. Essential Features of Parsons Turbine.

Westinghouse Machine Co., of Pittsburg, and by the Allis-Chalmers Co., of Milwaukee, Wis., the former acquiring the right to manufacture in 1885, and putting the first turbine on the market three years later. For marine purposes, licenses to build Parsons turbines have been issued to several firms.

The essential features of the Parsons turbine are clearly illustrated by the cross-section shown in Fig. 88. Steam enters at *E* and, in passing through the annular space between the cylinder walls and rotating elements, gradually expands in volume until it exhausts at *G*. The rotor is usually built in three different diameters to facilitate mechanical construction and to avoid excessively small and excessively large vanes. It is thus possible to use a large number of vanes of the same size. When the length of vane would otherwise become too great, the same passage area may be provided by shortening the vane and increasing the diameter of the drum. While, theoretically, the passage area of the vanes should gradually increase, it is found in practice that without any detrimental effect in economy several rows of vanes may be made of the same height, and thus the areas will increase step by step instead of in a gradual curve.

Since the pressure of steam is greater on the steam side of the vanes than on the exhaust side, there will result an end thrust which must in some way be balanced. This thrust, due to the static pressure of the steam, is augmented by the thrust on the vanes due to the impact and reaction of the steam in passing through them. To balance these thrusts, the balancing pistons shown at *L*, *M*, and *N*, Fig. 88, are provided. These are connected to the steam and exhaust spaces by the passages *O*, *P*, and *Q*, so that the pressure can be readily balanced. It is, of course, necessary to provide some sort of thrust block to meet the requirements of varying conditions, but it need not be large, and it serves in general as an adjustment bearing to keep the rotor in correct alignment, and is so arranged that the longitudinal position of the rotor may be slightly changed if desired. A mark on this block shows when the rotor is in its correct position.

Casing. The casing is made with diameters to accommodate the various sizes of drum and blading on the rotor. On the inside of the casing are the rings of fixed vanes or guides which fit between the rings of rotating vanes on the drum. The casing is divided horizontally, as shown in Fig. 89, so that by lifting the cover all working

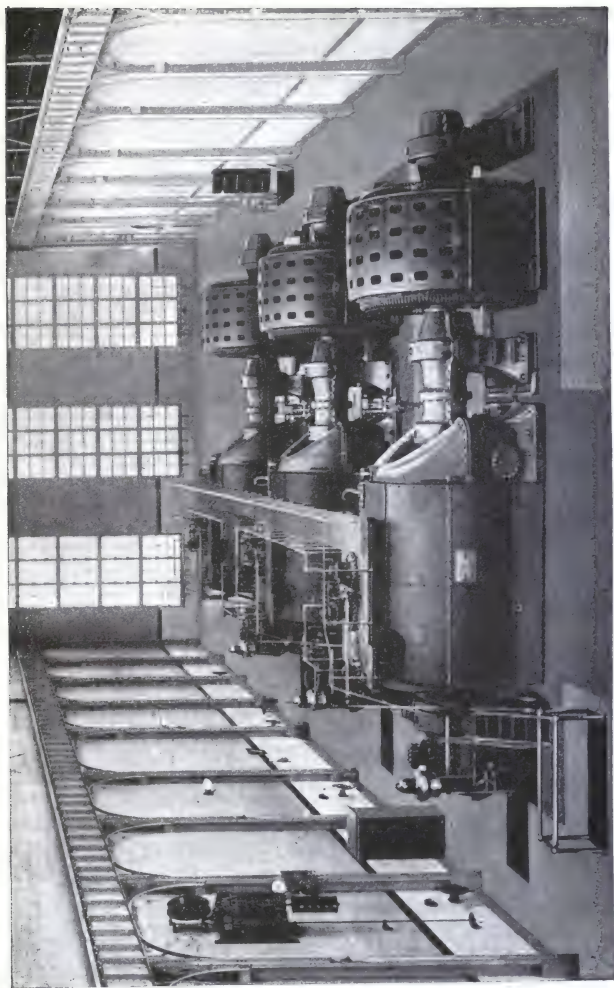


Fig. 89. Three Westinghouse-Parsons Steam Turbines Showing Divided Casing

parts are exposed. In very large turbines the cover slides over four graduated guides, one at each corner, so that in lifting it the engineer can readily see that it is kept always horizontal while being moved, without danger of binding or injuring the blading.

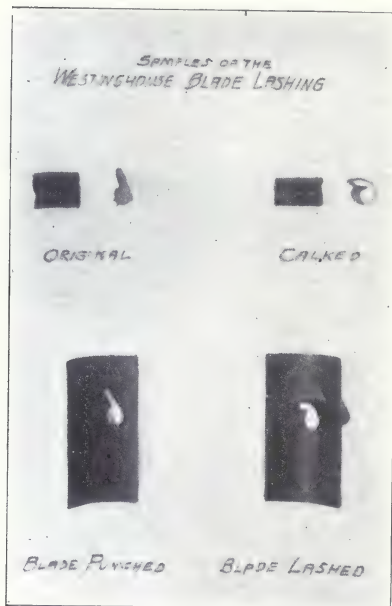


Fig. 90. Blade-Lashing of Westinghouse-Parsons Turbine.

Vanes. The vanes or blades, as they are most often called, are formed of special cold-drawn bronze or steel drawn out into strips and cut into proper lengths. Annular rings are turned on the outer side of the rotor and the inside of the casing, and in the grooves thus formed the blades are fitted and properly spaced by means of *distance pieces*, all carefully caulked in place. The entrance and exit angles of the vanes of the turbine not being equal, it would be an exceedingly difficult matter to properly set the blades were it not for these distance

pieces which insure not only the correct spacing but that the blade shall be set at the correct angle. To stiffen the outer ends of the blades and to maintain a uniform spacing, a wire lacing is threaded through openings near their outer edge, twisted between the adjacent blades, and soldered into position.

The Westinghouse Machine Co. use a cold-drawn wire lacing, or lashing of comma-shaped cross-section. This is threaded through similarly-shaped holes near the tips of the blades, and then the tail



Fig. 91. Rotor of Westinghouse-Parsons Turbine with Balance Pistons.

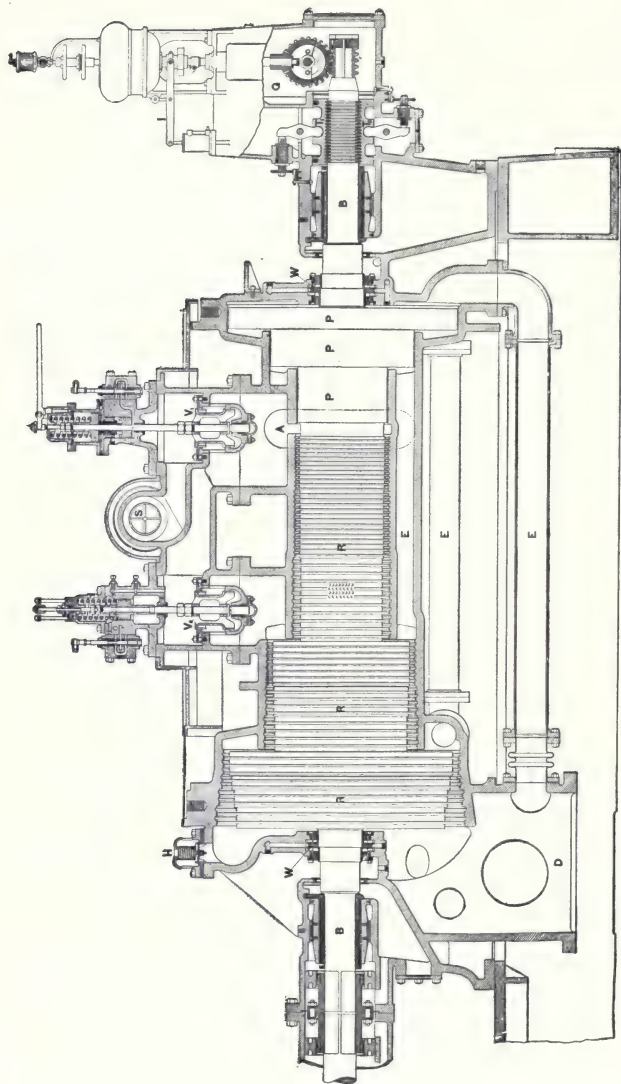


Fig. 92. Section of Westinghouse-Parsons Turbine.

of the comma is bent over between the blades, holding them firmly together, as shown in Fig. 90.

Rotor. The rotor, as before noted, consists of a drum of three diameters. This drum is usually built up of a hollow, steel casting fixed to some form of spider. The solid rotor would be prohibitively heavy. Fig. 91 shows the Westinghouse-Parsons rotor with balance pistons. Fig. 92 shows this turbine in cross-section.

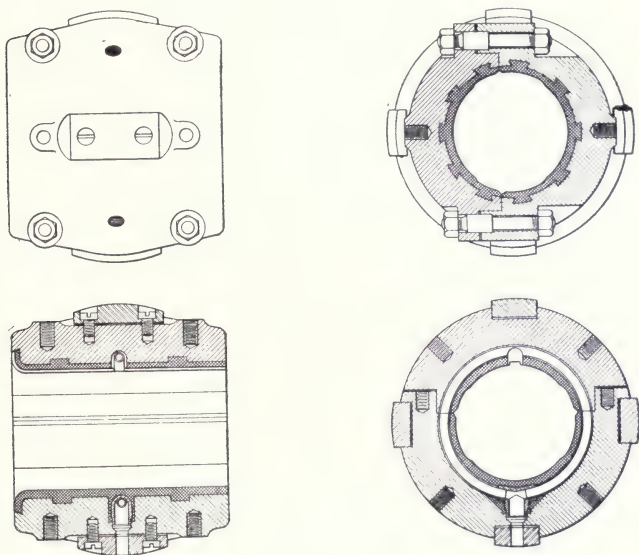


Fig. 93. Bearings of Westinghouse-Parsons Turbine.

Bearing. The rotor being long and heavy, some sort of flexible or self-aligning bearing is desirable, if not absolutely necessary. Flexibility is provided in the DeLaval turbine by means of the long, slender shaft, and in earlier forms of Parsons turbine by a nest of loosely-fitting concentric bronze bushings, with sufficient clearance to permit a continuous oil film between each two bushings. This was intended to form a cushion, permitting a certain amount of vibration in the shaft, yet restraining it within very narrow limits.

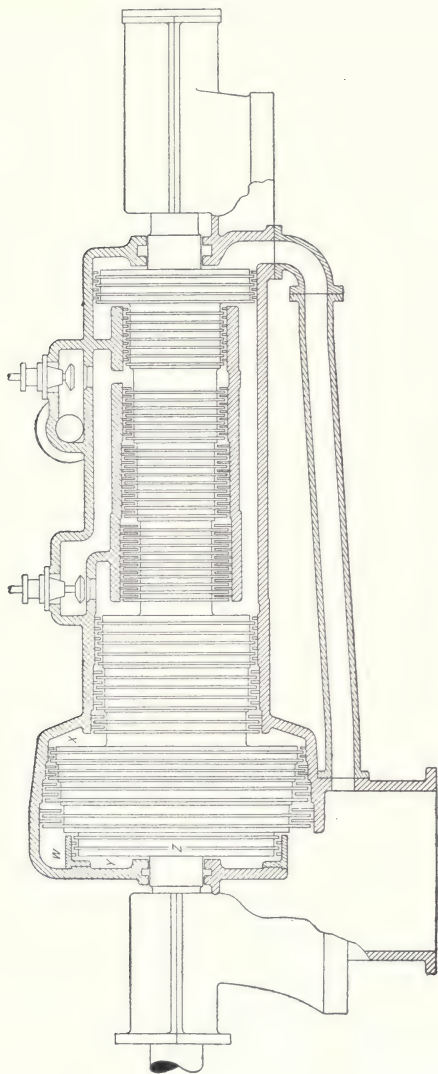


Fig. 94. Section of Allis-Chalmers Turbine.

It has been found by experience that this so-called flexible bearing is not necessary at the rotative speeds employed in Parsons turbines, and the Westinghouse Machine Co. have now adopted the style of bearing shown in Fig. 93. Turbines with short rotating shafts would not require a special form of bearing.

Allis-Chalmers Turbine. The turbine built by the Allis-Chalmers Co. of Milwaukee, is of the Parsons type with a few special modifications. It is claimed by its manufacturers that, in the large turbines, trouble is experienced with the largest or low-pressure balancing pistons. These being of comparatively large diameter, it is found difficult to so construct them that they will not be unduly distorted under pressure and repeated heatings and coolings. Distortion, of course, renders it impossible to run the packing rings on this piston with a sufficiently small clearance to prevent excessive leakage of steam. To overcome this difficulty, they have adopted the device shown in Fig. 94, which differs from the ordinary Parsons type shown in Fig. 88, by having the low-pressure balance piston placed at the low-pressure end of the turbine instead of at the high-pressure end, and making it of considerable less diameter. The ordinary Parsons piston has an effective area for balancing pressure equal to the difference between its area and the area of the next smaller piston; but, by putting this piston at the other end of the turbine, the whole circular area can be utilized and the same area may therefore be obtained with a much smaller diameter. This balancing piston is shown in *Z*, Fig. 94. It works inside of a supplementary cylinder *W*. In this construction, the equalizing pipe *P*, shown in Fig. 88, is omitted. The pressure on the balance piston *Z* is obtained by passages permitting communication between the steam spaces *X* and *Y*, but which are not shown on the figure. The balancing piston *Z* is not only much smaller than the piston in the ordinary type, but is stiffer, being backed up by the main body of the spindle.

In all reaction turbines, expansion of the casing is a troublesome feature; as this is due to the varying temperatures, there is an appreciable difference in the endwise expansion of the spindle and the casing. The high-pressure end of the spindle is held by a collar bearing, and the difference in expansion is taken up at the low-pressure end. The labyrinth packing employed at the high-pressure end has small axial and much radial clearance, while the labyrinth

packing of the balance piston at the low-pressure end may have small radial but must have large axial clearance to provide for the difference in expansion. The Allis-Chalmers Company claim that this type of construction permits smaller working clearances in high-pressure and intermediate pistons.

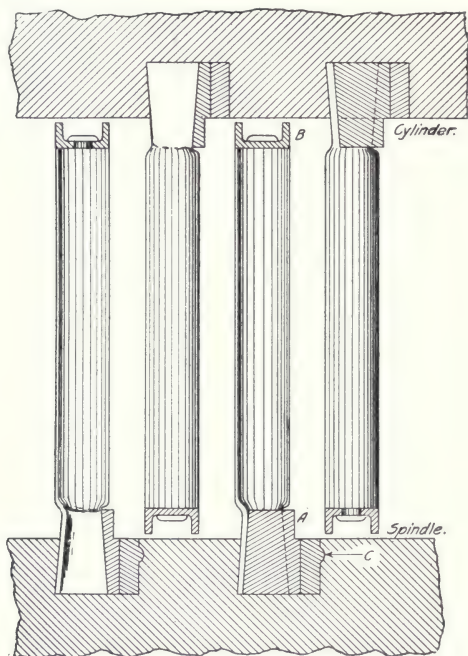
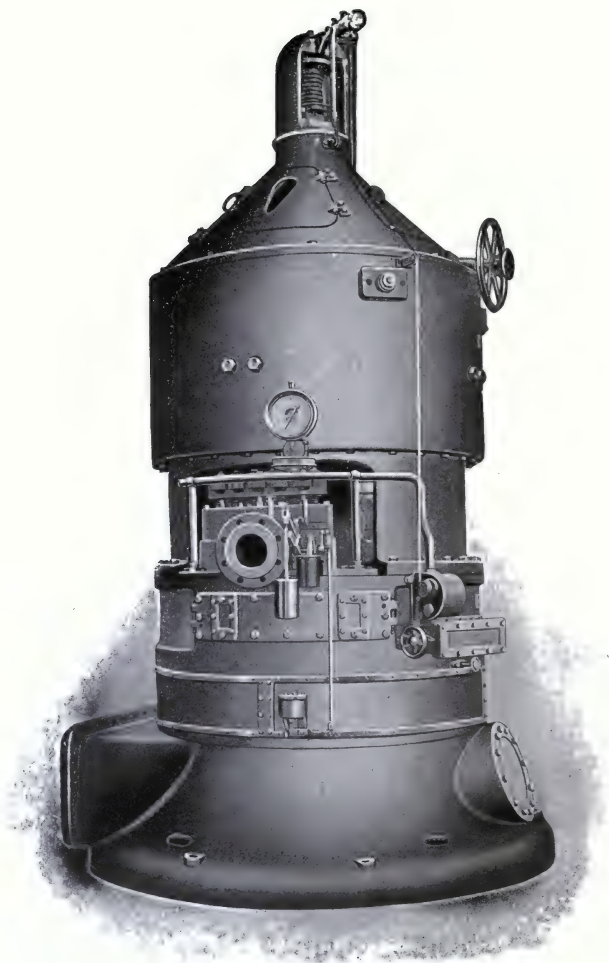


Fig. 95. Method of Fastening Blades of Allis-Chalmers Turbine.

Blading. The general shape of vanes of the Allis-Chalmers turbine does not differ from that of other reaction turbines, but the method of securing the blade to the casing and to the rotor is different from that adopted in the ordinary Parsons type. Each blade is so formed that, at its root, it is of an angular, dove-tail shape and has a small projection at its tip. To hold the roots of the blade firmly,



500 Kw. CURTIS STEAM TURBINE.
Direct Connected to Continuous Current Generator.

there is a foundation ring *A*, Fig. 95, which, after being formed to a circle of the proper diameter, has slots cut in it by a special milling machine, these slots being so shaped as to receive the roots of the blade. They are at the same time accurately spaced, and so cut as to give the required angle to the blades. To protect the tips of the blades and to bind them together in a substantial manner, a channel-shaped shroud ring *B*, Fig. 95, is fitted. The small projections on the tips of the blade fit through holes cut in this shroud ring and are riveted over. These rings, *A* and *B*, cover half the circumference and the blades are assembled on these rings, as shown in Fig. 96, before being put onto the rotor or casing. These foundation rings are of dove-tail shape in cross-section and are inserted into corresponding grooves in the turbine casing and spindle, in which they are firmly held by key pieces. These key pieces are driven into place and upset, so as to fill a small undercut, shown at *C* in Fig. 95, thus securely locking them into place. This construction is applied to all blades of whatever length.

The flanges of the channel-shaped, shroud rings are made thin, so that in case of contact with the casing from any accidental cause, no dangerous results are likely to follow, the accidental touch merely causing a slight wearing away of the flanges of the shroud without excessive heating. It is claimed that this shape of shroud ring acts in a measure like a labyrinth packing, retarding appreciably the leakage of steam. Fig. 97 shows a number of sets of blades as assembled.

GOVERNING

In order to regulate the supply of steam in proportion to the load on the steam turbine, there are several devices employed. (1) The pressure of the steam before admission to the steam chest may be controlled by means of a throttling governor. (2) In the case of impulse turbines with a series of nozzles, each may, if desired, be connected to an independent valve under the control of the governor, so that the number of open nozzles and the extent of the steam belt acting on the turbine may be varied to conform to the load, the steam being admitted to the nozzles at full pressure. (3) Steam may be admitted to the turbine at full pressure at intervals of long or short duration. Any of these devices may be arranged to take care of an ordinary amount of overload. For very large overloads, a by-pass

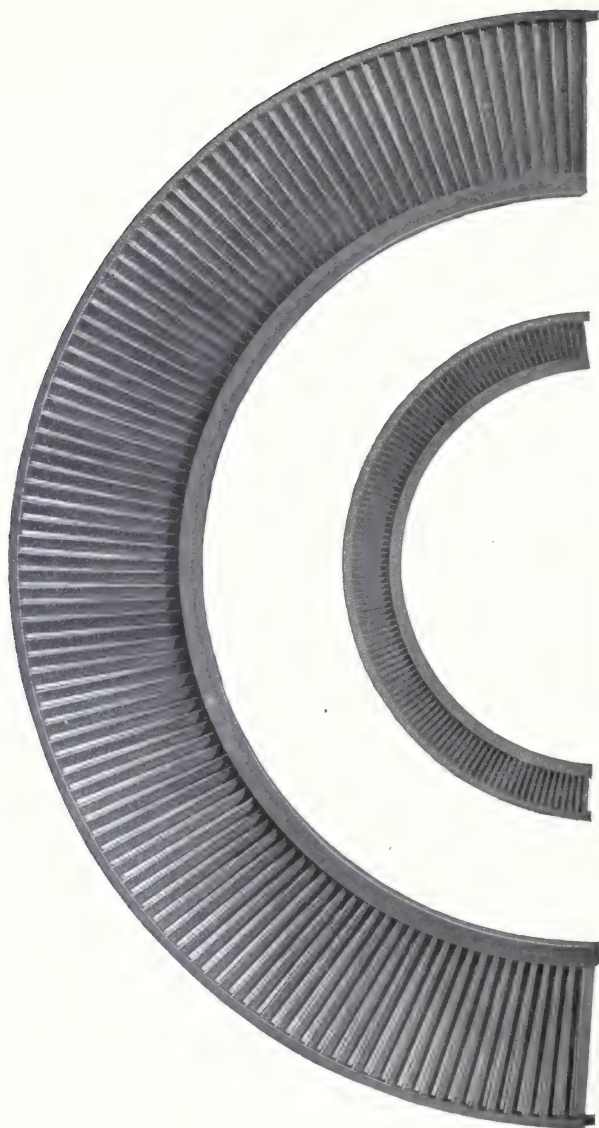


Fig. 96. Blades Assembled on Rings. *Allis-Chalmers Co.*

valve is generally provided, which may be controlled either by hand or by the governor, admitting high-pressure steam into one of the lower stages of the turbine.

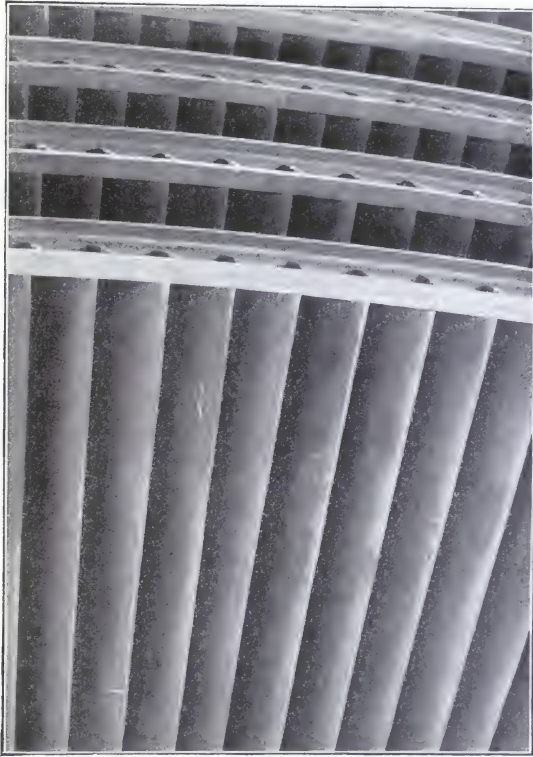


Fig. 97. Sets of Blades Assembled. *Allis-Chalmers Co.*

Of these devices, the second evidently is not applicable to the reaction turbine or to an impulse turbine taking steam around the entire periphery. The other methods of governing might be used for any form of turbine.

Throttling. The simplest governor is undoubtedly the throttle type, but throttling is not an economical way of regulating power; for, by the throttling process, the steam expands somewhat to a lower pressure without doing useful work. The work done goes to super-heat the steam and, although no heat units can be destroyed, less can be recovered from a pound of the throttled steam than from a pound at the initial pressure. The smaller the load, and consequently the more the throttling, the greater this loss will be. Any form of turbine, therefore, taking steam about its entire periphery and fitted with a throttling governor would probably be relatively uneconomical at light loads.

It is not economical to throttle the steam pressure before admission to a set of properly designed nozzles. Nozzles are designed for a definite steam pressure and, if this is varied, the efficiency is bound to fall off. There is, therefore, a two-fold loss due to throttling in a turbine of this type. In turbines like the DeLaval, supplied with several nozzles, any of which may be opened or closed by hand, it can readily be seen that if there are, for instance, four nozzles, three of which are open under ordinary loads, then a $33\frac{1}{3}\%$ or even $66\frac{2}{3}\%$ under-load may be taken care of by hand regulation. A throttling governor on such a turbine would necessarily regulate only between these limits and might act only upon one nozzle. On account of its simplicity, the throttling type of governor has been largely used and is practically always used on the smaller turbines where it has given results that are satisfactory.

Varying Number of Open Nozzles. Turbines employing partial admission through a group of nozzles like those of the Curtis turbine may regulate the steam supply by providing an independent valve for each nozzle. This valve, being under the control of the governor, may be opened or closed as needed. In the Curtis turbine, steam is admitted through a series of valves, the number of which depends upon the capacity of the machine. The valves are arranged to open successively, two-thirds of them being open at full load. The action of the valves is so regulated that they are either fully open or fully closed. Any increasing load is taken care of by the opening of an additional valve, this valve closing when the load falls off. Very wide variations in load can thus be carried with but little effect on steam economy. The friction, of course, is a very much larger

percent of small loads, so that the steam consumption at small loads would appear very much larger, even though it worked with the same absolute efficiency as at high loads.

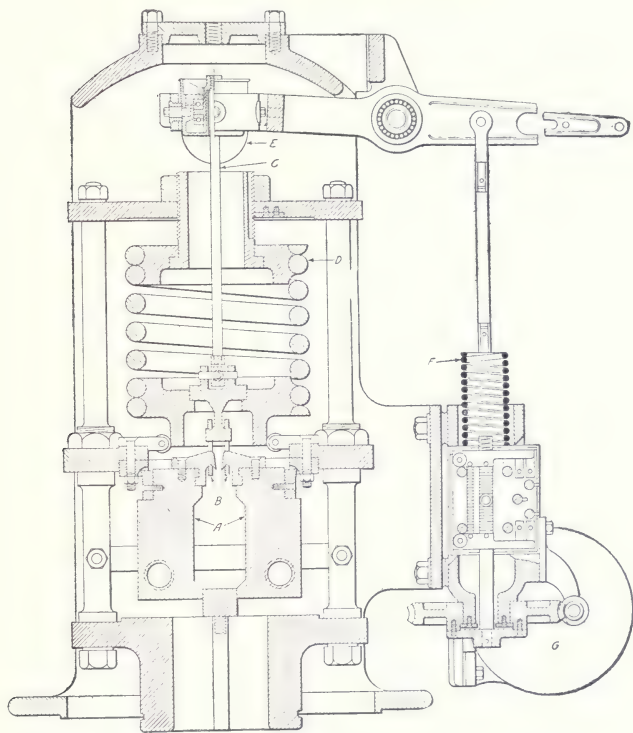


Fig. 98. Sectional View of Governor for Varying Number of Nozzles Opened.

These valves are controlled from the governor either by means of some electrical device, by a direct mechanical control, or by hydraulic pressure. The power necessary to operate one of these valves is necessarily so great that it cannot be moved directly by the governor, but the governor can operate a small pilot valve, which

in turn may set some mechanism in motion that is powerful enough to do the work.

The governor itself is of the centrifugal type, its action depending on the balance between the forces exerted by the springs and the centrifugal forces of the revolving weights. Fig. 98 shows a sectional view of this governor. *A* is the revolving weight, which, acting

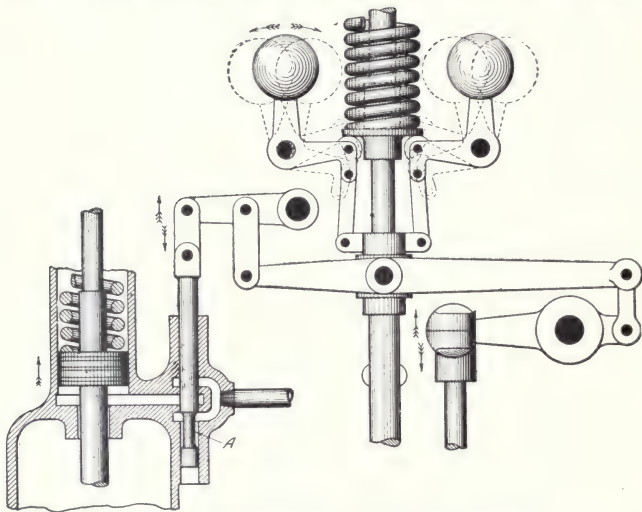


Fig. 99. Turbine Governor for Varying Time of Admission.

through the knife edges *B*, may move the rod *C* against the action of the spring *D*. At *E* is a ball-bearing gimbal joint which forms a junction between the revolving mass of the governor and the stationary lever of the governing arm. The governor is provided with the auxiliary spring *F* which may be used to vary the speed of the turbine. *G* is a small motor which may be operated from the switch-board to regulate the tension of this spring *F*, thus varying the speed of the turbine. The movement of the governor lever is transmitted through a connecting rod and lever to the pilot valve of an hydraulic cylinder. To provide for overloads greater than 50%, an auxiliary set of admission nozzles are provided in the second stage of the turbine, which may take full-pressure steam if needed.

Varying Time of Admission. The third method of governing that by varying the time of admission, is almost invariably used on turbines of the Parsons type. An admission of steam occurs about once in every thirty revolutions at approximately full load. The pilot valve is continually oscillating, thus preventing any liability of sticking, but its period of oscillation is varied directly by the governor. Steam, therefore, enters the turbine in puffs, the duration of which depends upon the load; at slight overloads the valve would be constantly open. One disadvantage of this method of governing is that there may be an opportunity for the turbine to cool between successive blasts of steam, thereby causing initial condensation at moderate and light loads. A by-pass valve, Fig. 94, is provided on Parsons turbines, and is under the direct control of the governor, and, for considerable overload, will open, letting steam into the intermediate stages.

The details of the mechanism for controlling steam turbine governors are not essentially different from those used in reciprocating engines, except as would be required by the greater speed of revolution. In turbines of large size, the valve cannot be operated directly by the governor, but must always be moved by means of pressure admitted through a small pilot valve. Fig. 99 shows the diagrammatic arrangement of the Westinghouse-Parsons governing gear. *A* is the pilot valve under the direct influence of the governor which admits steam to the larger piston, shown at the left. This piston has pressure enough upon it to operate the admission valves.

REVIEW QUESTIONS

PRACTICAL TEST QUESTIONS.

In the foregoing sections of this Cyclopedia numerous illustrative examples are worked out in detail in order to show the application of the various methods and principles. Accompanying these are samples for practice which will aid the reader in fixing the principles in mind.

In the following pages are given a large number of test questions and problems which afford a valuable means of testing the reader's knowledge of the subjects treated. They will be found excellent practice for those preparing for Civil Service Examinations. In some cases numerical answers are given as a further aid in this work.

REVIEW QUESTIONS

ON THE SUBJECT OF

HEAT.

EXERCISE I.

1. What is the difference between heat and temperature?
2. How are the freezing and boiling points of a thermometer determined?
3. To what temperature F does -10°C correspond?
4. To what temperature C does -20°F correspond?
5. What is the temperature of steam when the barometer stands at 30.05 inches?
6. What is the absolute zero of temperature?

EXERCISE II.

1. Give an example of linear expansion from your own observation.
2. A silver dollar is nicely fitted into an iron ring. If both are heated, will the coin fit more or less tightly?
3. A steel rail is 30 feet long at 20°C . How long is it at 0°C ?
4. A surveyor's steel tape 100 feet long is correct at 60°F . How much in error is it at 86°F ?
5. One end of every long iron bridge rests on rollers. Why is this?
6. 700 cubic feet of air at 20°C are heated to 50°C . Find the increase in volume.

HEAT

EXERCISE III.

1. What is the difference between evaporation and ebullition?
2. For what reason is it not difficult to boil water on top of a high mountain?
3. If a plug of tin were screwed into a steam boiler, would it remain in its position if the steam pressure were raised to 100 lbs. per square inch?
4. Upon what does the process of distillation depend? Why is the cooling water fed in at the bottom rather than at the top?
5. What is a British thermal unit?
6. What is the maximum possible efficiency of a steam engine taking steam at 185.3 pressure (gage) and exhausting at 15 pounds absolute pressure?

EXERCISE IV.

1. What is meant by the latent heat of fusion? Does the temperature of a substance change during solidification?
2. How many pounds of water at 60°C will be required to melt ten pounds of ice at -5°C ?
3. How many pounds of steam at 212°F will just melt 100 pounds of ice at 32°F ?
4. How many pounds of steam at atmospheric pressure will raise 2,000 pounds of water from 40°F to 50°F ?
5. Define specific heat in your own words.
6. What is superheated steam?

EXERCISE V.

1. Is there any thermal advantage in making a boiler of copper rather than of iron?
2. What is the difference between conduction and convection?
3. Suppose you wished to heat a liquid that contracted considerably on heating. Would you apply the heat at bottom or at top?

REVIEW QUESTIONS

ON THE SUBJECT OF

THE STEAM ENGINE.

PART I.

1. What should be the rim weight of a cast iron fly-wheel which is 15 feet in diameter, if the engine is of the Corliss type, with an 18" $\times$ 36" cylinder, and runs at 90 revolutions per minute?

2. In what way does the action of steam in the turbine engine differ from the action of steam in the ordinary reciprocating engine?

3. What is the theoretical height of a simple pendulum governor which makes 65 revolutions per minute?

4. In what way did Newcomen improve the steam engine?

5. Explain the difference between the condensing and the noncondensing engine, and show why the condenser increases the power of the engine.

6. Describe two methods of overcoming the danger due to high rotative speed in the steam turbine.

7. What two principles did James Watt follow in his experiments on the steam engine?

8. What types of engines have no fly-wheel? Why have they no fly-wheel?

9. What should be the thickness of rim for a cast iron fly-wheel weighing 4,480 pounds; the face is 12 inches wide, and the diameter of the wheel 15 feet?

10. Name some of Watt's inventions and improvements in the steam engine.

THE STEAM ENGINE.

11. What is meant by compound and triple expansion engines?
12. What is a Woolf engine? What is a tandem engine? A cross compound?
13. Explain with sketch the action of a sight-feed cylinder lubricator.
14. What is the function of a governor?
15. Why is not the power given out by an engine constant during a given time?
16. What are the relative advantages and disadvantages of turbine engines as compared with reciprocating engines?
17. What is the office of the fly-wheel?
18. How does the height of the governor affect its sensitivity?
19. Why are lubricants used? Name the requisites of good lubricants.
20. Name the advantages of the vertical engine.
21. Describe the two kinds of governors.
22. In what two ways do governors vary the work done on the piston?
23. How are high-speed engines oiled?
24. Which class of engine needs the larger fly-wheel, and why, high speed or low speed?
25. Describe the direct-acting steam pump.
26. How does the action of the Buckeye engine governor in changing the amount of steam admitted per stroke differ from the action of the straight-line engine governor?
27. What are the advantages and the principal disadvantages of the Corliss engine?
28. Name the requisites for high-speed reciprocating engines.
29. Why do pumping engines usually have devices for economy?
30. Explain how the expansion of steam is obtained in each of the two types of steam turbine.
31. If the height of a simple pendulum governor is 8 inches, what is the equivalent height if a weight equal to $1\frac{1}{2}$ times the weight of the balls is added?

REVIEW QUESTIONS

ON THE SUBJECT OF

THE STEAM ENGINE.

PART II.

1. Describe the method of finding the theoretical thermal efficiency.
2. Explain how there is any gain by using a steam jacket.
3. Explain the formula expressing the relation between pressure, volume and temperature of a perfect gas.
4. State Boyle's Law.
5. What effect does speed have on the economy of an engine?
6. Explain the advantages and disadvantages of superheating.
7. If steam exhausts into a jet condenser, how much water will be required (theoretically) to condense 10 pounds of steam, if the entering water is at 55° F, the discharge at 118° F and the vacuum gage reads 13.7?
8. At what position of the crank is the maximum turning moment?
9. If the feed water is heated, is the economy of the engine increased?
10. What is the absolute temperature of 150° F?
11. Other things being equal, is a large engine more economical than a small one? Why?
12. Explain initial condensation.
13. Describe the action of heat on a body of water under atmospheric pressure.

THE STEAM ENGINE.

14. Define the terms: saturated steam, superheated steam, wet steam and dry steam.

15. Name the properties found in the steam tables.

16. How does a condenser increase the power of an engine?

17. Explain how re-evaporation is possible.

18. How is the amount of steam used per hour found?

19. Explain with sketch how the crank effort is more nearly constant if there is more than one crank on the shaft.

20. In what two ways does compounding effect a saving of steam?

21. Give the formula expressing approximately the relation between pressure and volume of steam. What is it called?

22. Find the probable I. H. P. of a high-speed triple-expansion engine for a battle-ship. The engine runs at 150 revolutions; total ratio of expansion 12; initial pressure 145 pounds (gage), condenser vacuum 14 inches of mercury; the low-pressure cylinder is 72" $\times$ 36".

Ans. 2567.2 I. H. P.

NOTE.— Use card factor of .60 and atmospheric pressure 14.7 pounds.

23. What is the temperature of saturated steam corresponding to a pressure of 120 pounds (gage)?

24. What is the heat equivalent of work? How many foot-pounds of work equal one B. T. U.?

25. How many B. T. U. are required to furnish energy sufficient to raise 500 pounds 5,280 feet? How many pounds of coal will furnish this heat, assuming 6,800 B. T. U. per pound are available?

Ans. $\left\{ \begin{array}{l} 3,393 \text{ B. T. U.} \\ \frac{1}{2} \text{ pound (about).} \end{array} \right.$

26. How many cubic feet do 3 pounds of steam occupy when at 45 pounds absolute pressure?

27. How many B. T. U. are necessary to change 8 pounds of water at 320° F into steam at the corresponding temperature and pressure?

28. What is the actual thermal efficiency of an engine?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM ENGINE INDICATORS.

1. A Brumbo Pulley has the following dimensions: lever 80 inches long, stroke 32 inches and radius of sector 10 inches. What will be the length of the diagram?

2. Define Mean Effective Pressure.

3. When is a three-way cock used?

4. Describe Watt's Indicator.

5. A steam engine has a stroke of 30 inches, diameter of piston 18 inches, and the initial pressure 110 pounds (gage). If steam is admitted during the whole stroke and the exhaust is at atmospheric pressure, what is the theoretical M. E. P.?

6. In the above engine, what is the theoretical indicated horse-power if the engine is running at 75 revolutions per minute?

Ans. 318 I. H. P.

7. Why is a reducing motion necessary in taking indicator cards?

8. After taking an indicator card, how is the atmospheric line drawn?

9. Give the formula for finding the Indicated Horse-power, explain the meaning of the letters, and why is the denominator 33,000?

10. What is a good height and length for an indicator card? What data should be placed on the diagram?

11. Describe the process of taking a card.

INDICATORS.

12. Explain with sketch the method of finding the area of an indicator card by means of ordinates.

13. If you were told to find the I. H. P. of an engine what data would you require?

14. What peculiarity has the Tabor indicator?

15. What will be the fault with the card if the spring used in an indicator is too light? If too heavy?

16. If the I. H. P. is 85.6 and the efficiency 90 per cent, what is the available horse-power?

17. If the initial pressure of steam is 85 pounds absolute and the exhaust pressure 2 pounds above the vacuum, what is the thermal efficiency?

18. An indicator card gives a steam consumption of 28.6 pounds per I. H. P. per hour. If the mechanical efficiency is 87 per cent, what is the steam consumption per B. H. P.?

Ans. 32.87 pounds.

19. A rope brake is attached to the fly wheel of an engine. The pulley is 36 inches in diameter and the rope 1 inch. If the spring balance registers 220 pounds and the weight is 20 pounds, what is the B. H. P. while the engine is running 300 revolutions per minute?

Ans. 17.6 B. H. P.

20. Are the cards most reliable when taken by two indicators or when both are taken on the same paper by the same indicator?

21. What is the speed of an engine if the B. H. P. is 2.84, the length of arm of Prony brake 3 feet, and the weight at the end of the arm 35 pounds?

Ans. 142 revolutions.

22. If the admission line of an indicator diagram curves backward, what does it indicate? How would you remedy it?

23. What is wire-drawing? Explain why it is undesirable.

24. The piston of an engine is 19 inches in diameter; the piston speed is 480 feet per minute. A card was taken with a 40-pound spring; the mean ordinate being 1.16 inches. What was the I. H. P.?

Ans. 191.4 I. H. P.

25. What causes wavy lines in an indicator diagram?

26. What is the indicated horse-power of an engine $12'' \times 18''$ when running at 90 revolutions, the M. E. P. being 42 pounds?

REVIEW QUESTIONS

ON THE SUBJECT OF

VALVE GEARS.

1. Why cannot the displacement of the piston be found in the same manner as that of the valve?

2. What is the throw of the eccentric if the valve travel is 3 inches and the rocker arm has the following dimensions: $AB = 12$ inches, $AG = 15$ inches. See Figs. 19 and 20.

3. Explain why the eccentric radius is not at right angles to the crank. What is lead?

4. Why are laps used? What effect does an increase of outside lap have upon steam distribution?

5. Describe an eccentric.

6. Sketch a plain slide valve and indicate ports, bridges, exhaust cavity, inside and outside lap.

7. Give the advantages of the Corliss valve gear.

8. With a connecting rod of ordinary length, explain why there is unequal steam distribution, if the valve is set for one end and if the laps are equal.

9. How do the following changes affect steam distribution with a plain slide valve. *a.* Increase in angular advance. *b.* Increase in eccentricity.

10. Describe the essential features of the Joy valve gear.

11. Referring to Fig. 16, will the line RO be perpendicular to XY when the piston is in the middle of the stroke?

12. What should be the diameter of the steam pipe for an $18'' \times 24''$ engine that makes 100 revolutions per minute?

Ans. 5 inches.

13. Describe how the governor regulates the engine by means of the shifting eccentric.

14. When setting the Meyer valve, is the main valve given a late or early cut-off? Why?

15. What relation between cut-off and compression can be obtained with the Meyer valve that is impossible with the ordinary slide valve?

16. Describe, with sketch, the Stephenson link.

17. Explain why an engine runs in the opposite direction if the valve is disconnected from one eccentric and connected to the opposite one.

18. What should be the width of the exhaust port?

19. What is the advantage of the piston valve?

20. What are the advantages of the radial valve gears?

21. What is one disadvantage of the Corliss gear?

22. How may a valve be most easily balanced to overcome weight of valve and gear as well as steam pressure?

23. How does the Gooch link differ from the Stephenson link?

24. Given in a certain valve:

Valve travel = 3 inches.

Lead angle = 6° .

Cut-off at $\frac{5}{8}$ stroke.

Compression at $\frac{3}{4}$ stroke.

Ratio of connecting rod to crank = 4.

Find by means of Zeuner's diagram:

Linear lead.

Outside lap.

Inside lap.

Angular advance.

25. What are "crossed rods"? How is steam distribution with crossed rods different from that with open rods?

26. How does the passage in the trick valve affect lead? How does it affect admission?

27. For what two reasons are link motions used on locomotives?

28. Why does bringing the block nearer the center of the Stephenson link cause cut-off to occur earlier?

29. What advantage has the double-ported slide valve?

30. What is meant by the mid-position of a valve?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM TURBINES

PART I

1. How does the steam economy of a good steam turbine compare with that of the best Corliss engine of the same power?
2. What is the most serious difficulty that confronts the steam turbine designer, and how may it be overcome?
3. What is the difference between the *impulse* and *reaction* types of turbine?
4. In an impulse turbine designed for a given pressure and vacuum, explain fully what effect on its economy would be produced by changes in boiler pressure.
5. How does a nozzle giving maximum velocity to a steam jet differ from one giving maximum velocity to a water jet, and why?
6. How does the steam expansion in the DeLaval differ from that in the Rateau turbine?
7. Why are comparisons of performance based on steam consumption per H. P. per hour, unreliable?
8. Why does no commercial turbine act by virtue of pure reaction only?
9. How should a nozzle expanding steam at maximum efficiency from 200 lbs. absolute to 120 lbs. absolute differ from one expanding it from 200 lbs. absolute to atmospheric pressure?
10. Explain the function of a Rateau accumulator?
11. What advantage is to be gained by compounding a steam turbine? Why?

STEAM TURBINES

12. What are the chief sources of loss in the steam turbine?
13. What are the essential requirements in the design of a steam turbine?
14. What is the most serious objection to the compound velocity step turbine, if the blade angles are all equal?
15. How are large overloads provided for in steam turbines?
16. If a nozzle makes an angle of 20° with the plane of rotation, what is the proper peripheral velocity of the vanes for maximum efficiency if the jet velocity is 4,000 feet per second?
17. In any reaction turbine what chief factor limits the economy?
18. What is the most reliable method of comparing the performance of two steam engines or turbines?
19. What is a so-called *low-pressure* steam turbine?
20. If a cylindrical nozzle delivers steam from a pressure of 100 lbs. absolute into a pressure of 58 lbs. absolute, how would the energy of the jet be affected by raising the initial pressure to 200 lbs.? By lowering the final pressure to atmosphere?
21. How does the cost of a complete turbine power plant including buildings compare with that of a reciprocating engine plant of equal power?
22. What is the relation between the number of pressure stages in a multi-stage machine and the reduction in the number of revolutions?
23. How does the method of converting the heat units of steam into useful work in the turbine differ from that in the reciprocating engine?
24. Explain in what way the problem of designing a steam turbine differs from that in the case of a water turbine.
25. How does the action of steam in the Rateau turbine differ from that in the Curtis turbine?
26. What effect does a *variation of load* have on the steam economy of a turbine?
27. Explain why the reciprocating engine cannot make economical use of the lower ranges of vacuum.
28. If a turbine is compounded by velocity steps only, what relation is there between the speed reduction and the number of velocity steps?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM TURBINES

PART II

1. In what ways may steam turbines be compounded?
2. What is meant by *single stage*? *four stage*? Name the principal turbines of the multi-stage group.
3. To what class of turbines does the Rateau belong? the Westinghouse-Parsons? the Terry?
4. Describe the method of governing a Curtis turbine.
5. Explain a *labyrinth packing*. What is a *shroud ring*?
6. How were rotative speeds reduced in the Riedler-Stumpf single-stage machine? How, in the DeLaval?
7. What turbines belong to the compound-velocity step group?
8. Describe the bearings of a Curtis vertical turbine.
9. In the Zoelly turbine how are the revolutions and centrifugal stresses kept within moderate limits?
10. What device is employed on the DeLaval turbine to counteract any slight lack of balance of the rotor? How does this differ from that employed by the Parsons designers?
11. Explain fully why no other turbines are provided with such devices.
12. What is the usual method of lubrication on rotor shaft bearings? Why is ring oiling unsatisfactory for rotor bearings of the DeLaval turbine?
13. What is the function of a *by-pass* and how does the by-pass operate on a Curtis turbine?
14. How does the DeLaval turbine differ from the Rateau?
15. In what way do the turbines of the compound-velocity step group differ from one another?

STEAM TURBINES

16. In what particulars does the Allis-Chalmers turbine differ from the Westinghouse?
17. How does the Terry two-stage turbine differ from a two-stage Curtis?
18. What devices have been employed to prevent leakage from stage to stage in the multi-stage group?
19. Explain the action of steam in the Curtis turbine.
20. What device is adopted on the DeLaval turbine to prevent wrecking the entire machine if it approached a dangerous speed limit?
21. How did the steam action in the Riedler-Stumpf differ from that in the Curtis?
22. What distinctive features has the Kerr turbine?
23. How does the Zoelly turbine differ from others of its group?
24. What are the distinctive features of the Curtis turbine?
25. What disadvantages are connected with the DeLaval method of reducing speeds of rotation?
26. To what group does the Curtis belong? What others belong to the same group?
27. What is the most important factor in the economy of a reactive turbine?
28. How does the rotor of a Parsons turbine differ from the rotor of a Rateau?
29. How do the turbines of the multi-stage group differ one from the other in the matter of steam expansion?
30. What are the different methods of governing a turbine?

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